Flame-Wall Interaction of a Flame Jet Impinging Normally on a Cooled Cylinder

By

Maikel van der Steen

Master Thesis

Submitted in Partial Fulfillment of the Requirements for the Master of Science in Applied Physics at the Delft University of Technology

Delft, The Netherlands
2014
Acknowledgments

Throughout this master thesis I have received help from several people.

I would like to thank my advisor Dr.ir. M.J. Tummers who proposed this research to me. I would like to thank him for the freedom I got in designing and performing the experiments. Setting up a new experiment is not common for a Master student and I would like to thank him for that opportunity. I have always felt comfortably inside the group and I have really enjoyed my time here. The discussions we had always helped me to move in the right direction.

I would also like to thank Ing. Bart Hoek, who was always prepared to help me and give me technical advice. Especially in the building phase of the experiment he had very useful input. Whenever I needed some components he would get these for me or whenever I needed some help with building, he would help me. The expertise he had acquired over the years really helped me along.

I would like to thank Ing. Erwin de Beus for all ICT support he gave me. When I needed help finding the right DAQ equipment or had any problems with the laser, he was there to help me.

I would also like to thank the visitor from the Czech Republic, Dr. Jiri Vondal. We worked together closely for a couple of months. Especially in the building phase of the experiment it really speeded things up.

I would also like to thank the PhD students Milos Birvalski, Luis Arteaga Mendez, Hugo Rodriguez and Rudi Kalter for their occasional help on various subjects. They were always prepared to give me advice, even though they were very busy working on their own project.

Finally I would like to thank my parents for their support all these years.
ABSTRACT

The main objective of the present experimental investigation is to study the turbulent reacting flow field in the region where a stable premixed flame jet (Re = 3250 and Φ = 1.3) impinges normally on a cooled cylinder and gain a better understanding of the effect of the cylinder wall temperature. PIV measurements were done in a plane normal to the cylinder axis for both a cold wall (100 °C) and a hot wall (500 °C). The effect of wall temperature on mean flow velocities and Reynolds stresses were investigated by comparing the results of detailed PIV measurements for both cases.

The effect of wall temperature on the mean flow velocities is minimal. In the impinging jet region, small differences in the mean velocities only exist near the wall due to thermal expansion. In the wall jet region, small differences in the mean velocities arise further away from the wall caused by the mean position where large scale vortices are created. It was found that the position where these vortices form are generally further upstream for a hot wall than for a cold wall. Large Reynolds stresses are found at two locations: (i) At the location where the inner flame bends around the wall, large Reynolds stresses are present due to the wavy motion of the inner flame as a result of shear between the fast unreacted cold core and the slow diffusion flame. (ii) At the outer edges of the diffusion flame, large Reynolds stresses are present due to the periodic passing of large scale vortices, altering the velocity flow field in the wall jet region. At both locations the Reynolds stresses increase with increasing cylinder wall temperature, which is related to the position where vortices form.
Table of Contents

1 Introduction ......................................................... 1
  1.1 Background of the Investigation ............................... 1
  1.2 Objectives of the Investigation ............................... 2
  1.3 Outline of the Thesis ......................................... 2

2 Turbulent Flows ...................................................... 4
  2.1 Basics of Turbulent Flows ..................................... 4
    2.1.1 Equations of Fluid Motion ............................... 4
    2.1.2 Favre-Averaged Navier-Stokes Equations ................. 5
  2.2 Combustion ..................................................... 7
    2.2.1 Flame Type ............................................ 7
    2.2.2 Flame Structure ....................................... 7
    2.2.3 Equivalence Ratio ..................................... 7
    2.2.4 Laminar Burning Velocity .............................. 8
    2.2.5 Flame Front .......................................... 8

3 Measurement Techniques ........................................... 10
  3.1 Particle Image Velocimetry .................................. 10
    3.1.1 Introduction .......................................... 10
    3.1.2 Equipment ........................................... 11
      High-Speed Laser .......................................... 11
      Optics .................................................. 11
      Seeding ................................................. 12
      High-Speed Camera ....................................... 13
      Wall Reflection .......................................... 14
3.1.3 Analysis .................................................. 14
   Cross-Correlation ....................................... 14
   Criteria ................................................. 15
3.2 Thermocouples ............................................. 16

4 Experimental Setup ........................................... 18
4.1 Burner Configuration ........................................ 18
   4.1.1 Burner Geometry ..................................... 18
   4.1.2 Flame Power ........................................ 20
4.2 Cylinder Wall .............................................. 20
   4.2.1 Cylinder Geometry ................................... 20
   4.2.2 Thermocouples ...................................... 21
   4.2.3 Temperature Profile .................................. 22
4.3 Cooling Configuration ...................................... 24
4.4 PIV Setup ................................................ 25
   4.4.1 Laser ................................................ 25
   4.4.2 Laser Optics ........................................ 25
   4.4.3 Tracer Particles .................................... 26
   4.4.4 Camera .............................................. 26
4.5 Safety Precautions .......................................... 27
   4.5.1 Ultra Violet Sensor .................................. 27
   4.5.2 Shielding ............................................ 27

5 Results ....................................................... 28
5.1 Flame Appearance .......................................... 28
   5.1.1 Stability Map ....................................... 28
   5.1.2 Visual Appearance of the Flames .................. 29
5.2 Mean Flow and Reynolds Stresses .......................... 31
   5.2.1 Impinging Flame Jet Region ....................... 32
   5.2.2 Wall Jet Region ...................................... 35
5.3 Effect of Wall Temperature ................................ 42
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$D$</td>
<td>cylinder outer diameter</td>
<td>m</td>
</tr>
<tr>
<td>$D_s$</td>
<td>section diameter</td>
<td>m</td>
</tr>
<tr>
<td>$H$</td>
<td>distance between burner pipe and cylinder wall</td>
<td>m</td>
</tr>
<tr>
<td>$Kw_s$</td>
<td>friction coefficient</td>
<td>-</td>
</tr>
<tr>
<td>$L$</td>
<td>characteristic length</td>
<td>m</td>
</tr>
<tr>
<td>$L_s$</td>
<td>section length</td>
<td>m</td>
</tr>
<tr>
<td>$M$</td>
<td>magnification of the camera</td>
<td>-</td>
</tr>
<tr>
<td>$N$</td>
<td>particles per interrogation area</td>
<td>-</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
<td>-</td>
</tr>
<tr>
<td>$Nu_c$</td>
<td>air jet Nusselt number</td>
<td>-</td>
</tr>
<tr>
<td>$Nu_h$</td>
<td>flame jet Nusselt number</td>
<td>-</td>
</tr>
<tr>
<td>$P_{\text{flame}}$</td>
<td>flame power</td>
<td>W</td>
</tr>
<tr>
<td>$Q$</td>
<td>volumetric flow rate</td>
<td>m$^3$ s$^{-1}$</td>
</tr>
<tr>
<td>$Q_{ht}$</td>
<td>heat transfer rate</td>
<td>W</td>
</tr>
<tr>
<td>$Re$</td>
<td>burner pipe exit Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$Re_c$</td>
<td>air jet Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$Re_h$</td>
<td>flame jet Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>$S$</td>
<td>burning velocity</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$S_L$</td>
<td>laminar burning velocity</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$S_T$</td>
<td>turbulent burning velocity</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$Stk$</td>
<td>Stokes number</td>
<td>-</td>
</tr>
<tr>
<td>$S_{ij}$</td>
<td>rate of strain tensor</td>
<td>s$^{-1}$</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
<td>K</td>
</tr>
<tr>
<td>$T$</td>
<td>total time</td>
<td>s</td>
</tr>
<tr>
<td>$T_c$</td>
<td>air jet temperature</td>
<td>K</td>
</tr>
<tr>
<td>$T_h$</td>
<td>flame jet temperature</td>
<td>K</td>
</tr>
<tr>
<td>$T_{wc}$</td>
<td>cylinder inner wall temperature</td>
<td>K</td>
</tr>
<tr>
<td>$T_{wh}$</td>
<td>cylinder outer wall temperature</td>
<td>K</td>
</tr>
<tr>
<td>$V$</td>
<td>voltage</td>
<td>V</td>
</tr>
<tr>
<td>$V_{\text{seebeck}}$</td>
<td>Seebeck voltage</td>
<td>V</td>
</tr>
<tr>
<td>$V_b$</td>
<td>bulk velocity</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$d$</td>
<td>burner pipe inner diameter</td>
<td>m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
<td>------</td>
</tr>
<tr>
<td>$d_e$</td>
<td>particle image diameter</td>
<td>m</td>
</tr>
<tr>
<td>$d_p$</td>
<td>tracer particle diameter</td>
<td>m</td>
</tr>
<tr>
<td>$d_s$</td>
<td>Airy function diameter</td>
<td>m</td>
</tr>
<tr>
<td>$e_{fr}$</td>
<td>energy dissipation by friction</td>
<td>m$^2$ s$^{-2}$</td>
</tr>
<tr>
<td>$f$</td>
<td>focal length</td>
<td>m</td>
</tr>
<tr>
<td>$f_s$</td>
<td>section fanning friction factor</td>
<td>-</td>
</tr>
<tr>
<td>$f^#$</td>
<td>f-number of the lens</td>
<td>-</td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient</td>
<td>W m$^{-2}$ K$^{-1}$</td>
</tr>
<tr>
<td>$k$</td>
<td>turbulence kinetic energy per unit mass</td>
<td>m$^2$ s$^{-2}$</td>
</tr>
<tr>
<td>$k_c$</td>
<td>thermal conductivity of the cold air</td>
<td>W m$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$k_{dw}$</td>
<td>density weighted turbulence kinetic energy per unit mass</td>
<td>kg m$^{-1}$ s$^{-2}$</td>
</tr>
<tr>
<td>$k_h$</td>
<td>thermal conductivity of the hot flame jet mixture</td>
<td>W m$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$k_{th}$</td>
<td>thermal conductivity</td>
<td>W m$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$k_w$</td>
<td>thermal conductivity of the cylinder wall</td>
<td>W m$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$p$</td>
<td>static pressure</td>
<td>N m$^{-2}$</td>
</tr>
<tr>
<td>$p'$</td>
<td>fluctuating static pressure</td>
<td>N m$^{-2}$</td>
</tr>
<tr>
<td>$\bar{p}$</td>
<td>mean static pressure</td>
<td>N m$^{-2}$</td>
</tr>
<tr>
<td>$q''$</td>
<td>heat flux</td>
<td>W m$^{-2}$</td>
</tr>
<tr>
<td>$r$</td>
<td>radial distance from centre of cylinder wall</td>
<td>m</td>
</tr>
<tr>
<td>$\vec{s}$</td>
<td>displacement vector</td>
<td>m</td>
</tr>
<tr>
<td>$t$</td>
<td>thickness cylinder wall</td>
<td>m</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
<td>s</td>
</tr>
<tr>
<td>$u_i$</td>
<td>instantaneous velocity in the $x_i$-direction</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$u_i'$</td>
<td>fluctuating velocity in $x_i$-direction</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$u_i''$</td>
<td>density weighted fluctuating velocity in $x_i$-direction</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$\vec{v}$</td>
<td>velocity vector</td>
<td>m</td>
</tr>
<tr>
<td>$x_i$</td>
<td>position</td>
<td>m</td>
</tr>
<tr>
<td>$\bar{u}_i$</td>
<td>mean velocity in $x_i$-direction</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$\hat{u}_i$</td>
<td>density weighted mean velocity in $x_i$-direction</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$\frac{\vec{u}_i \cdot \vec{u}_j}{u_i u_j}$</td>
<td>Reynolds stress tensor</td>
<td>m$^2$ s$^{-2}$</td>
</tr>
<tr>
<td>$&lt; v &gt;_s^2$</td>
<td>section mean flow speed</td>
<td>m s$^{-1}$</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>pressure drop</td>
<td>Pa</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>pulse delay time</td>
<td>s</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>temperature difference</td>
<td>K</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>equivalence ratio</td>
<td>-</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>angle in cylindrical coordinate system w.r.t. centreline</td>
<td>rad</td>
</tr>
<tr>
<td>$\epsilon_{ij}$</td>
<td>viscous dissipation rate</td>
<td>kg m$^{-1}$ s$^{-3}$</td>
</tr>
<tr>
<td>$\delta_{ij}$</td>
<td>Kronecker delta function</td>
<td>-</td>
</tr>
<tr>
<td>$\delta$</td>
<td>unburned layer thickness</td>
<td>m</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>wavelength</td>
<td>m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>-------</td>
<td>-------------------------------------------------------</td>
<td>----------------</td>
</tr>
<tr>
<td>$\mu$</td>
<td>dynamic viscosity</td>
<td>$\text{kg m}^{-1} \text{s}^{-1}$</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
<td>$\text{m}^2 \text{s}^{-1}$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density of the fluid</td>
<td>$\text{kg m}^{-3}$</td>
</tr>
<tr>
<td>$\rho_p$</td>
<td>density of the particle</td>
<td>$\text{kg m}^{-3}$</td>
</tr>
<tr>
<td>$\rho u_i u_j$</td>
<td>density weighted Reynolds stress tensor</td>
<td>$\text{kg m}^{-1} \text{s}^{-2}$</td>
</tr>
<tr>
<td>$\sigma_{ij}$</td>
<td>stress tensor (in Newtonian fluid)</td>
<td>$\text{kg m}^{-1} \text{s}^{-2}$</td>
</tr>
<tr>
<td>$\tau_f$</td>
<td>timescale of the flow</td>
<td>$\text{s}$</td>
</tr>
<tr>
<td>$\tau_p$</td>
<td>characteristic time of the particle</td>
<td>$\text{s}$</td>
</tr>
</tbody>
</table>
# List of Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>CCD</td>
<td>Charge-Coupled Device</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data AQuisition</td>
</tr>
<tr>
<td>DNG</td>
<td>Dutch Natural Gas</td>
</tr>
<tr>
<td>FANS</td>
<td>Favre-Averaged Navier-Stokes</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transformation</td>
</tr>
<tr>
<td>FT</td>
<td>Fourier Transformed</td>
</tr>
<tr>
<td>LDA</td>
<td>Laser Doppler Anemometry</td>
</tr>
<tr>
<td>MFC</td>
<td>Mass Flow Controller</td>
</tr>
<tr>
<td>Nd:YAG</td>
<td>Neodymium-doped Yttrium Aluminum Garnet</td>
</tr>
<tr>
<td>Nd:YLF</td>
<td>Neodymium-doped Yttrium Lithium Fluoride</td>
</tr>
<tr>
<td>NS</td>
<td>Navier-Stokes</td>
</tr>
<tr>
<td>PIV</td>
<td>Particle Image Velocimetry</td>
</tr>
<tr>
<td>RANS</td>
<td>Reynolds-Averaged Navier-Stokes</td>
</tr>
</tbody>
</table>
CHAPTER 1

Introduction

1.1 Background of the Investigation

The interaction between a flame and a wall is an important topic in both combustion and heat transfer research. Flame-wall interaction may lead to important modifications of the flame and the flow dynamics.

Flame-wall interaction in combustion equipment is often an unwanted effect. It can have adverse effects on the overall conversion and the pollutant formation of the flame. It can also decrease the lifetime of the wall. In combustion equipment, the wall often functions as a cooling device. Experimental observations show that the flame strength decreases near cold wall surfaces due to the increased wall heat flux, leading to a significant drop of the flame temperature to a value below its ignition temperature (Gruber and Chen, 2010). This phenomenon is called quenching and has been investigated extensively. Two types of quenching configurations can be distinguished. When a flame quenches while traveling parallel to a wall it is called sidewall quenching. When a flame quenches when normally impinging on a wall it is called head-on quenching. One of the first works that investigated this quenching layer was done by Daniel (1957). This led to a growing number of investigations that focused on combustion engines due to the practical use in the design of internal combustion engines (Lu and Sawyer, 1991; Ezekoye and Sawyer, 1992; Vosen and Westbrook, 1985; Kurkov and Mirsky, 1969; Lavoie and Blumberg, 1980; Ferguson and Keck, 1977). The main goal of these investigations was to get a better insight in the processes that affect the pollutant formation, efficiency and lifetime of the engines. In these investigations quenching occurred primarily at the sidewalls and not head-on. Due to the flame-wall interaction taking place in a very thin layer, early experiments only focused on the measurement of the unsteady wall heat flux. The maximum wall heat flux decreased with increasing wall temperature due to a decreased temperature gradient between the flame and wall as shown by Ezekoye and Sawyer (1992). Eventually, improvements in measurement techniques enabled the direct measurement of the quenching layer. Enomoto (2002) did this using a shadowgraph method and photographs of a high-speed movie camera. Increased computational power enabled the study of complex direct numerical simulations (Gruber and Chen, 2010).

In most practical applications where combustion is used for intense heating, the flame is turbulent. Turbulence generally increases the shear stresses and the heat transfer
in the near wall region such that very thin velocity and thermal boundary layers are formed, especially at higher Reynolds numbers. Accurate measurements of the velocity and temperature in this thin boundary layers are very challenging. Impinging flame jets as opposed to non-reacting impinging jets also adds complexity to the flow. Flames change the heat transfer and quenching can occur in the vicinity of the wall. Continuous improvements in the spatial and temporal resolution of measurement techniques slowly shift the relative amount of investigations from laminar to turbulent flows. Measurements in the vicinity of the wall are still very difficult due to the difficult optical access and spurious scattering when using techniques such as Particle Image Velocimetry (PIV) and Laser Doppler Anemometry (LDA).

1.2 Objectives of the Investigation

Flame-wall interaction in internal combustion engines has been studied extensively in the last few decades. Most of these investigations focused on laminar flows due to limitations of the measurement techniques. Only in the last decade the relative amount of turbulent flows has become a significant contribution to flame-wall interaction studies. Intense heating applications, where the flame is turbulent, have been studied much less than internal combustion engines and leaves many gaps to be filled. The temperature of the wall can vary significantly over time and influences the lifetime of the wall and possible also the flow dynamics. The main objective of the present experimental investigation is therefore to study the reacting turbulent flow field in the region where a premixed flame jet impinges on a cooled cylinder. The wall temperature on the reacting flow is studied by varying the temperature of the cylinder wall.

To capture the flame-wall interaction in detail, PIV is used. PIV is a technique that relies on the use of high-speed CCD camera’s. The size of the turbulent structures and the speed of the velocity fluctuations that can be captured are limited by the current camera and laser capability.

1.3 Outline of the Thesis

Chapter 2 is concerned with the basics of turbulent flows and combustion. Section 2.1 focuses on the tools that are used in turbulent reacting flows, especially the Reynolds-averaged equations of motion for compressible turbulent flows. Section 2.2 briefly summarizes on some properties of combustion such as the types of flames, structure of the flames, equivalence ratio, burning velocity and formation of the flame front.

Chapter 3 is devoted to the measurement techniques that are used. Section 3.1 focuses on PIV. Section 3.2 focuses on thermocouples.
Chapter 4 discusses the experimental setup. The experimental setup is described in parts where in Section 4.1 the burner configuration is discussed, in Section 4.2 the cylinder wall, in Section 4.3 the cooling configuration, in Section 4.4 the PIV setup and in Section 4.5 the safety precautions.

Chapter 5 presents and discusses the results of the investigation. In Section 5.1 the stability and appearance of the flames are discussed. The mean velocities and Reynolds stresses of a specific flame are investigated in Section 5.2. Finally, the effect of wall temperature is investigated in Section 5.3.

Finally, in Chapter 6 the conclusions of the investigation are given.
2.1 Basics of Turbulent Flows

In this chapter the tools that are used in turbulent combustion flow are briefly studied. The derivations will not go in great depth as only the parts essential to understanding the later chapters will be discussed. For a more detailed discussion the reader is referred to a textbook such as the one by Pope (2000). In the last part, the equivalence ratio and burning velocity will be discussed shortly as they are essential to understanding the investigation.

2.1.1 Equations of Fluid Motion

The equations of fluid motion for turbulent combustion flows consist of the compressible Navier-Stokes (NS) equations and the compressible continuity equation. These equations can be derived from the conservation of momentum and mass. The conservation of mass for a compressible fluid reads

\[ \frac{\partial \rho}{\partial t} + \frac{\partial \rho u_i}{\partial x_i} = 0, \quad (2.1) \]

where \( \rho \) is the density and \( u_i \) the instantaneous velocity component in the \( x_i \) direction. The Einstein notation is used in this equation to sum over all values of \( i \). The conservation of momentum of a compressible fluid is expressed by

\[ \frac{\partial \rho u_i}{\partial t} + \frac{\partial \rho u_i u_j}{\partial x_j} = \frac{\partial \sigma_{ij}}{\partial x_j}. \quad (2.2) \]

The terms on the left hand side represents the time rate of change of \( u_i \). It is the acceleration of momentum and the advection of momentum. The term on the right hand side represents the forces acting on the fluid parcel. In case of a Newtonian fluid, the stress tensor \( \sigma_{ij} \) can be expressed as

\[ \sigma_{ij} = -p\delta_{ij} + 2\mu S_{ij}, \quad (2.3) \]
where \( p \) is the pressure containing the hydrodynamic pressure, \( \delta_{ij} \) the Kronecker delta, \( \mu \) the dynamic viscosity and \( S_{ij} \) the strain rate tensor given by

\[
S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{1}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij}.
\]  

(2.4)

Combining Eqs. 2.2 and 2.3 results in the compressible NS equations

\[
\frac{\partial p}{\partial t} + \frac{\partial p u_i}{\partial x_i} = - \frac{\partial p}{\partial x_i} + \mu \left[ \frac{\partial}{\partial x_j} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \frac{\partial u_k}{\partial x_i} \frac{\partial u_k}{\partial x_k} \right].
\]  

(2.5)

The compressible NS equations (Eq 2.5) together with the compressible continuity equation (Eq. 2.1) and the boundary conditions describe the turbulent fluid motion in detail spatially and temporally. The compressible NS equations describe how a change in momentum of a fluid parcel is in equilibrium with pressure gradients and dissipative viscous forces working on the parcel. The compressible NS equations model this exactly. A problem however resides in the solution to these equations, since no exact solutions can be found. Direct numerical simulations can be a good approximation to an exact solution but for high Reynolds numbers and complex geometries they require great amounts of computational power. Experimental data in combination with numerical solutions of the Favre-Averaged Navier-Stokes equations (FANS equations) are used to be able to model the fluid motions.

### 2.1.2 Favre-Averaged Navier-Stokes Equations

The FANS equations are a set of equations holding the density weighted time averaged equations of fluid motion. To obtain the FANS equations, a technique called Reynolds decomposition is used. In Reynolds decomposition, the instantaneous quantity is decomposed into a time-averaged part and a fluctuating part. In the case of the NS equation, the pressure and instantaneous velocities are subject to Reynolds decomposition:

\[
u_i = \overline{u_i} + u_i' \quad \text{and} \quad p = \overline{p} + p',
\]

(2.6)

where the overbar expresses the time-averaged quantity and the prime denotes the fluctuating part. The time-averaged quantity described by the Reynolds-averaged is expressed as

\[
\overline{u_i} = \lim_{T \to \infty} \frac{1}{T} \int_0^T u_i(t) dt.
\]

(2.7)

Since the flow is compressible, the velocity fluctuations may cause fluctuations in the density, therefore a density weighted average is introduced where the decomposition is expressed as
\[
\hat{u}_i = \hat{u}_i + u_i'',
\]

where the hat denotes the density weighted average expressed as
\[
\hat{u}_i = \lim_{T \to \infty} \frac{\int_T^T \rho(t)u_i(t)dt}{\int_T^T \rho(t)dt} = \frac{\overline{\rho u_i}}{\overline{\rho}}.
\]

In most cases Favre averaging is not required, since turbulent fluctuations most often do not lead to any significant fluctuations in density. Applying Reynolds averaging to the compressible continuity equation and the compressible NS equation will yield the averaged continuity equation for a compressible flow and the FANS equations

\[
\frac{\partial \rho}{\partial t} + \frac{\partial \rho \hat{u}_i}{\partial x_i} = 0,
\]

\[
\frac{\partial \rho \hat{u}_i}{\partial t} + \frac{\partial \rho \hat{u}_i \hat{u}_j}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \mu \left[ \frac{\partial}{\partial x_j} \left( \frac{\partial \hat{u}_i}{\partial x_j} + \frac{\partial \hat{u}_j}{\partial x_i} \right) - \frac{2}{3} \frac{\partial \hat{u}_k}{\partial x_i} \right] - \frac{\partial \rho u_i'' u_j''}{\partial x_j}.
\]

The last term stands out in this equation as it is the only term that is not present in the compressible NS equation (Eq 2.5). These velocity correlations \(\rho u_i'' u_j''\) resides from the non-linear term of Eq. 2.5. As a result of Reynolds averaging, information is lost which is now contained in the velocity correlation term. The quantity \(\rho u_i'' u_j''\) is known as the Reynolds-stress tensor and is denoted by \(\tau_{ij}\). It has six independent components that next to the three mean velocity components \(\overline{u_i}\) and the mean pressure \(\overline{p}\) are needed to close the equation. Generally, the mean velocity and mean pressure components are easy to obtain. The Reynolds stress tensor however is hard to obtain and causes the set of equations to no longer be closed. This is called the closure problem of the averaging approach. The Reynolds-stress tensor has to be expressed with the means of ‘turbulence modeling’.

A popular model in numerical simulations is the \(k-\epsilon\) turbulence model. The density weighted version of \(k\) is the density weighted turbulence kinetic energy per unit mass \(k_{dw}\), defined as
\[
k_{dw} = \frac{1}{2} \rho u_i'' u_i''.
\]

The \(k-\epsilon\) model will not be described as it is not inside the scope of this investigation. However a detailed description can be found in Pope (2000).
2.2 Combustion

2.2.1 Flame Type

To classify combustion phenomena it is useful to introduce two types of flames: diffusion flames and premixed flames. In a diffusion flame, fuel from the burner mixes by convection and diffusion with the ambient oxidizer before combustion. In a premixed flame, fuel and air are already mixed within the burner. The gas is ignited and the premixed flame front is found at a steady state position.

2.2.2 Flame Structure

A premixed flame can be sub-categorized in a single-flame, double-flame and a triple-flame. Single, double and triple refers to the number of regions present in a flame. Only single-flames and double-flames that are possible without co-flow are discussed. When there is co-flow, there are many more possibilities. This is not covered here, however it is discussed in detail in Hou and Lin (2003).

A single-flame is either fuel-lean or very fuel-rich. In a lean single-flame all reactions occur in a single region because all fuel reacts quickly after it leaves the burner. A very fuel-rich single-flame has too little oxidizer to form a stable flame front without external oxidizer. It only reacts after mixing with ambient oxidizer to form a diffusion flame.

A double-flame is a fuel-rich flame consisting of two regions. It has a fuel-rich inner flame surrounded by a diffusion flame. In the inner flame most fuel reacts with the premixed oxidizer. Remaining fuel and reaction products form a diffusion flame after reacting with ambient oxidizer.

2.2.3 Equivalence Ratio

The equivalence ratio of a fuel-oxidizer mixture describes the mass ratio of fuel to oxidizer, described by

\[ \Phi = \frac{(m_{\text{Fuel}}/m_{\text{Oxidant}})_{\text{actual}}}{(m_{\text{Fuel}}/m_{\text{Oxidant}})_{\text{stoichiometric}}} \]  

(2.13)

Here, \( \Phi = 1.0 \) is defined as the stoichiometric condition where after reaction both fuel and oxidizer have fully reacted. When there is an excess of fuel the mixture is called ‘rich’ and \( \Phi > 1.0 \). When there is an excess of oxidant present the mixture is called ‘lean’ and \( \Phi < 1.0 \).
2.2.4 Laminar Burning Velocity

The burning velocity $S$ is a measure of the rate at which the reactants move towards the flame observed within a reference frame at the flame front. The burning velocity is a function of the equivalence ratio. For a methane-air mixture at atmospheric pressure and room temperature the laminar burning as a function of the equivalence ratio is shown in Fig. 2.1 (Dahoe and de Goey, 2003). The maximum burning velocity is found around $\Phi = 1.0$. Lower and higher equivalence ratios result in lower burning velocities. The maximum burning velocity depends on the temperature, the pressure and the turbulence intensity. Velocity fluctuations in the turbulent regime increase the burning velocity. The ratio of turbulence burning velocity to laminar burning velocity $S_T/S_L$ can have a value as high as 10 (Shy and Wei, 2000), increasing the burning velocity of a methane-air mixture to well over 4 m/s for high Reynolds numbers.

![Fig. 2.1](image)

Fig. 2.1: The laminar burning velocity $S_L$ as a function of the equivalence ratio $\Phi$ for a methane-air mixture at atmospheric pressure and room temperature (Dahoe and de Goey, 2003).

2.2.5 Flame Front

The steady state position of a flame front is such that the components of the velocity vectors normal to the flame front are equal to the burning velocity. The position of the flame front depends on the flow rate and the burning velocity. Three cases can be distinguished.

When the flow rate is smaller than the burning velocity, the flame propagates upstream until there is no more fuel or it is extinguished by a flame arrestor. This is called flashback.
When the flow rate is slightly larger than (or equal to) the burning velocity, in case of a laminar flow, the flame front will be conical shaped due to the shape of the laminar velocity profile. Increasing the Reynolds number causes velocity fluctuations in the flow field. This results in a steady state position of the flame front that varies in time. The mean position of the flame front has the shape of the burner pipe exit flow velocity profile.

When the flow rate is significantly larger than the burning velocity, no steady state position can be found and the flame will propagate upstream until all fuel has reacted. This is called blowoff.
CHAPTER 3
Measurement Techniques

3.1 Particle Image Velocimetry

3.1.1 Introduction

Particle Image Velocimetry (PIV) is an optical technique used to measure fluid velocity in a planar cross section of the flow. A typical PIV setup is shown in Fig. 3.1 (Raffel and Kompenhans, 2007). The flow is seeded with small particles that accurately follow the flow. The seeding particles are illuminated twice by a laser light sheet. A series of cylindrical lenses is used to transform the laser beam into a light sheet. The images of seeding particles are recorded on the CCD of a high-speed camera. Two sequential images are used to determine the particle displacement $s$ by using cross-correlation techniques. The particle velocity $\vec{v}$ follows from the displacement and the time between two laser pulses $\Delta t = (t' - t)$ simply as $\vec{v} = s/\Delta t$.

Fig. 3.1: A typical PIV setup (Raffel and Kompenhans, 2007).

The main advantage of PIV is that it measures the flow field non-intrusively. Measurement techniques using a physical probe such as hot wire anemometry disturb the flow.
Another advantage of PIV is its ability to do velocity measurements in the whole plane. Probes and other techniques such as Laser Doppler Anemometry (LDA) can usually only measure at a single point. PIV has a reasonable spatial resolution (but less than LDA) and a high temporal resolution. Both the temporal and spatial resolution of PIV are continuously improving with the development of better high-speed camera’s and lasers.

3.1.2 Equipment

High-Speed Laser

The most commonly used lasers in PIV are the frequency doubled Nd:YAG lasers and the frequency doubled Nd:YLF lasers. Nd:YLF lasers have a higher pulse energy at a low pulse rate. For PIV the lasers are of dual oscillator/single head type which means that there are two lasers inside the head and the light beams of these lasers are combined by precision internal optics. Double cavity lasers are used because the time between consecutive pulses is often very small (in the order of \( \mu s \)). The lasers are high repetition rate, diode pumped lasers, which allows for complete control of pulse delay time \( \Delta t \) and pulse energy. The Nd:YLF lasers have a pulse-width in the order of 200 ns and a maximum pulse repetition rate in the order of 10 kHz. The small pulse-width is needed to prevent motion blur in the images of the tracer particles. The small pulse delay time \( \Delta t \) can be generated by firing laser A and B right after each other as shown in Fig. 3.2. The repetition rate of the pulse pairs is then equal to the repetition rate of the individual lasers.

![Small delay pulse pairs.](image)

Optics

A series of cylindrical lenses are used to form a planar laser light sheet. There are multiple possibilities to form such a sheet. Fig. 3.3 shows a frequently used configuration. First, a plano-concave cylindrical lens is used to vertically diverge the laser beam. Then a plano-convex cylindrical lens converts it into a thick parallel planar laser light sheet.
Finally a second plano-convex cylindrical lens is used to converge the thick planar laser light sheet horizontally into a thin planar laser light sheet.

Fig. 3.3: Laser optics.

Seeding

PIV relies on images of illuminated tracer particles. These tracer particles are commonly referred to as ‘seeding’. A main requirement for seeding particles is their ability to follow flow fluctuations accurately. The Stokes number describes the degree at which the particles follow the flow and is defined as

\[ Stk = \frac{\tau_p}{\tau_f}, \]

where \( \tau_p \) is the characteristic time of the particle and \( \tau_f \) the timescale of the flow. The characteristic time of the particle \( \tau_p \) is the time interval during which the particle accelerates from zero velocity up to the value of 63.2\% of the liquid velocity and can be described (for Stokes flow) as

\[ \tau_p = \frac{\rho_p d_p^2}{18\mu}, \]

where \( \rho_p \) is the particle density, \( d_p \) the particle diameter and \( \mu \) the dynamic viscosity. For \( Stk > 1 \) the particles detach from the flow. For \( Stk < 1 \) the particles follow the flow closely. A small Stokes number requires a small particle diameter \( d_p \).

On the other hand, the particle diameter \( d_p \) should be large enough to scatter enough light to be visible for the high-speed camera. This results in a trade-off between these two requirements. The particles should be small enough not to alter the fluid or flow properties, yet large enough to reflect enough laser light. A number of different particles that can be used as tracer particles are listed in Table 3.1 for gaseous flows (Raffel and Kompenhans, 2007). In gaseous flows with high Reynolds number the particle diameter \( d_p \) is in the micrometre range.
<table>
<thead>
<tr>
<th>Type</th>
<th>Material</th>
<th>$d_p$ (µm)</th>
<th>$\tau_p$ (µs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solid</td>
<td>Polystyrene</td>
<td>0.5 - 10</td>
<td>0.9 - 378</td>
</tr>
<tr>
<td></td>
<td>Alumina $\text{Al}_2\text{O}_3$</td>
<td>0.2 - 5</td>
<td>0.6 - 378</td>
</tr>
<tr>
<td></td>
<td>Titania $\text{TiO}_2$</td>
<td>0.1 - 5</td>
<td>0.15 - 378</td>
</tr>
<tr>
<td></td>
<td>Dioctylphthalate</td>
<td>1 - 10</td>
<td>3.4 - 378</td>
</tr>
<tr>
<td></td>
<td>Smoke</td>
<td>&lt; 1</td>
<td>&lt; 0.004 0</td>
</tr>
<tr>
<td>Liquid</td>
<td>Different oils</td>
<td>0.5 - 10</td>
<td>0.8 - 302</td>
</tr>
<tr>
<td></td>
<td>Di-ethyl-hexyl-sebacate (DEHS)</td>
<td>0.5 - 1.5</td>
<td>0.9 - 7.7</td>
</tr>
</tbody>
</table>

Table 3.1: Gaseous flow tracer particles.

High-Speed Camera

To determine a single instantaneous velocity field with PIV, two sequential exposures of the particles are required. For high-speed PIV, the camera frame rate for its maximum resolution can go up to 10 kHz. This is enough to resolve the largest timescales in many laboratory scale combustion experiments where the flow speed is not very high.

When small delay pulse pairs are captured as shown in Fig. 3.2, the camera exposure has to be timed perfectly. This is done with the help of an external (digital) synchronizer. The synchronizer sets the beginning of an exposure right between the two pulses and ends it between the next two as shown in Fig. 3.4. The high-speed camera framerate is twice the pulse repetition rate.

![Fig. 3.4: Exposures.](image)

The size of the particle image is determined by the particle diameter $d_p$, the magnification of the camera $M$ (image size to object size ratio) and the point response function of the lens. If the particles are diffraction limited, the point response function of the lens can be expressed as an Airy function of diameter

$$d_s = 2.44(1 + M)f^\#\lambda,$$

where $\lambda$ is the wavelength of the laser light and $f^\#$ is the f-number of the lens describing the ratio of the lens’s focal length to the diameter of the entrance pupil. The image captured by the camera is a convolution of the Airy function with the geometric image of the particle. If both functions are approximated with a Gaussian, the following
approximate formula for the image diameter can be found (Adrian, 1991):

\[ d_e = \sqrt{M^2 d_p^2 + d_s^2}. \]  

(3.4)

For the most common high-speed camera setups used in PIV, \( d_e \) can be approximated by \( d_s \) for particle diameters \( d_p \) smaller than 10\,\mu m.

Wall Reflection

Diffuse surface reflections of laser light are often a concern in PIV measurements. Surface reflections can result in local overexposure of the CCD in the near-surface area. This generally compromises the quality of the cross-correlations in the region of the flow near the walls. There are various measures that can be taken to minimize or even completely remove any surface reflections. Possible strategies to achieve this can be divided in four categories: (i) Absorbing the beam energy. By using an absorbing material for the wall, the reflected intensity can be reduced. (ii) Maximizing the surface reflection in a direction away from the camera and minimizing the surface reflection in the direction of the camera. This can for example be done by polishing the surface and positioning the camera parallel to the surface. (iii) Maximizing the transmittance of the surface. This can be done by using transmitting materials such as glass. (iv) Fluorescent seeding or paint can be used in combination with a bandpass filter. Fluorescent seeding can be used to absorb the laser light and emit at another wavelength. By using a bandpass filter isolating the emitted wavelength, the fluorescent light can be captured. It is also possible to use fluorescent paint on the wall in combination with a bandpass filter in front of the camera isolating the laser light wavelength. In combustion experiments fluorescent paint or seeding is not an option. Fluorescent paint or particles are only heat resistant up to about 200 °C, where in combustion experiments temperatures are hundreds of degrees higher.

After minimizing the wall reflection as far as possible, it is possible to remove the remaining surface reflection by masking it or using image post-processing. In some cases this might be succesful, but the risk of removing important data is great.

3.1.3 Analysis

Cross-Correlation

The field of view is subdivided into a number of small regions called interrogation areas. In an interrogation area, the velocity of the flow \( \vec{v} = \frac{\vec{s}}{\Delta t} \) is assumed to be uniform. The time interval between two sequential exposures \( \Delta t \) should be such that this requirement is met. Each interrogation area of the first image is correlated with the same interrogation area of the second image. A cross-correlation of an interrogation area of two sequential exposures using a Fast Fourier Transform (FFT) is illustrated in
Fig. 3.5. The interrogation area of both exposures are Fourier Transformed (FT) after which the FT image of the first exposure ($\hat{I}_1$) is multiplied with the conjugate of the FT image of the second exposure ($\hat{I}_2^*$). The result ($\hat{I}_1 \cdot \hat{I}_2^*$) is then inverse transformed (FT$^{-1}$) to obtain the displacement vector $\vec{s}$.

Fig. 3.5: Schematic representation of a cross correlation of interrogation areas between two sequential exposures (Raffel and Kompenhans, 2007).

Criteria

The error of the cross-correlations increase with increasing particle density as $1/\sqrt{N}$ where $N$ is the number of particles in each interrogation area. A rule of thumb is that the density of tracer particles should be such that there are at least ten particles for each interrogation area for a good cross-correlation. Fig. 3.6 shows the result of a cross-correlation for 5, 10 and 25 particles per interrogation area.

Fig. 3.6: Cross-correlation for $N = 5$, 10 and 25 (Raffel and Kompenhans, 2007).

In-plane displacements and out-of-plane displacements have adverse effects on the accuracy of the cross-correlations. In-plane displacements are when particles moves from one interrogation area to another in between exposures. In order to obtain a good cross-correlation, the displacement of the particles in the direction of the plane should be less than a quarter of the interrogation area. This limits the size of the interrogation areas. A way to overcome this is to use the multi-pass approach. In this approach, cross-correlations are done on larger interrogation areas which show the direction velocity of the flow. Next, smaller interrogation areas are shifted in that direction between the first and second image resulting in almost no in-plane loss.
Out-of-plane displacements are when particles move out of the laser sheet. For an accurate cross-correlation, the displacement of the particles in between exposures in the direction away from the laser sheet should be less than a quarter of the laser sheet thickness.

The optimal particle diameter is a trade-off between bias errors that occur for particle diameters smaller than 1 pixel and random noise errors for diameters larger than 2 pixels. The optical particle diameter has a size of 2 pixels (Westerweel, 1997).

### 3.2 Thermocouples

Temperature cannot be measured as distance or time is measured. Temperature must be related to a measurable physical phenomena. In a typical mercury thermometer for example, the expansion of the mercury is related to temperature. In a thermocouple it is somewhat similar as a voltage difference is related to temperature.

If two dissimilar metals are joined at both ends and some temperature gradient is applied to it, a continuous current flows in this circuit which is called a ‘thermoelectric circuit’. This discovery was made by Thomas Seebeck in 1821. Separating the ends at the non-measurement side, results in a voltage difference between these two ends called the ‘Seebeck Voltage’ as illustrated in Fig. 3.7. For small changes in temperature, the Seebeck Voltage is linearly proportional to temperature.

![Fig. 3.7: Seebeck Voltage due to a temperature gradient over two dissimilar wires.](image)

In order to measure this voltage difference, a voltage meter needs to be attached to the two ends. However, this voltage meter creates a thermoelectric circuit itself. The wires of the voltage meter are of different material than the thermocouple wires, creating a Seebeck Voltage between them as well. The measured voltage $V$ therefore is not just the Seebeck Voltage from the thermocouple wires but also the Seebeck Voltages between the thermocouple wires and the voltage meter wires $V_2$ and $V_3$ added to them as shown in Fig. 3.8.

The Seebeck Voltage over the thermocouple wires $V_1$, is thus expressed as

$$ V_1 = V - V_2 - V_3. \quad (3.5) $$

The junctions between thermocouple wires and voltage meter wires are thermally isolated by a isothermal block and their temperatures are measured by a thermistor inside
the isothermal block. By knowing the material of the wires, the Seebeck Voltages $V_2$ and $V_3$ can be read out.

Over the years, some combinations of alloys have proven themselves and have become popular. Factors such as the cost, availability, melting point, chemical properties and accuracy play a part in this. The two most important parameters are the temperature range and the sensitivity. A small list of some of the most common thermocouples have been listed in Table 3.2.

<table>
<thead>
<tr>
<th>Type</th>
<th>Material A</th>
<th>Material B</th>
<th>Sensitivity ($\mu$m/$^\circ$C)</th>
<th>Temperature range ($^\circ$C)</th>
<th>Curie point ($^\circ$C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E</td>
<td>Chromel</td>
<td>Constantan</td>
<td>68</td>
<td>-50 to 740</td>
<td>Outside range</td>
</tr>
<tr>
<td>J</td>
<td>Iron</td>
<td>Constantan</td>
<td>50</td>
<td>-40 to 750</td>
<td>770</td>
</tr>
<tr>
<td>K</td>
<td>Chromel</td>
<td>Alumel</td>
<td>41</td>
<td>-200 to 1350</td>
<td>350</td>
</tr>
<tr>
<td>N</td>
<td>Nicrosil</td>
<td>Nisil</td>
<td>39</td>
<td>-270 to 1300</td>
<td>Outside range</td>
</tr>
<tr>
<td>T</td>
<td>Copper</td>
<td>Constantan</td>
<td>43</td>
<td>-250 to 350</td>
<td>Outside range</td>
</tr>
</tbody>
</table>

Table 3.2: Some of the most common thermocouple types and their characteristics.

The Curie point is an important temperature at which the material’s permanent magnetism changes to induced magnetism. In a thermocouple this causes a smooth change in the characteristics which will result in a deviation in the output around this temperature. In some cases this will set the upper limit for the temperature range. In thermocouple type J for example, if the material reaches a temperature above 760 $^\circ$C this will result in an abrupt magnetic transformation that causes permanent decalibration. The sensitivity is expressed in ($\mu$m/$^\circ$C) and will determine the accuracy of the measurement in combination with the accuracy of the voltage meter.
Chapter 4
Experimental Setup

In this chapter the experimental setup is discussed. It was build and designed by ourselves, therefore this chapter will not only discuss the basic parts of the experimental setup, but also the design decisions and underlying calculations. In Fig. 4.1, a graphical representation of the experimental setup is shown. Most important are the burner pipe with inner diameter \( d \) and length 55d and a hollow steel cylinder with outer diameter \( D \) positioned directly above it. From the burner pipe exit a premixed Dutch Natural Gas (DNG)/air flame jet will impact on the cylinder wall. The DNG/air ratio or equivalence ratio \( \Phi \) can be set with flow meters and valves. To ensure the mixture flow is straight, a honeycomb is inserted in the burner pipe. Internal cooling of the cylinder wall with thickness \( t \) induces a temperature gradient. Cooling is provided by an impinging (ambient temperature) air jet coming through a hole in the top of the cylinder and produced by a side channel blower. Inside the cylinder, ten miniature J-type thermocouples are point welded to measure the wall temperature. The flow of the flame is measured by PIV.

In the next few chapters the experimental setup is described in more detail. First, the burner configuration including the ultraviolet sensor is discussed in Chapter 4.1. Then in Section 4.2 the cylinder wall is discussed including the thermocouples and the data acquisition hardware. The cooling configuration supplying the impinging air jet is discussed in Section 4.3. In Section 4.4 the PIV configuration is discussed. Finally in Section 4.5 the safety precautions are discussed.

4.1 Burner Configuration

4.1.1 Burner Geometry

The burner pipe is made of copper and has an inner diameter \( d = 26.0 \) mm and a wall thickness of \( 1.0 \) mm. The pipe has a length of 1.43 m which corresponds to 55d. A honeycomb has been inserted three quarters upstream to ensure there is no swirl in the flow resulting from the sharp junction corner at the upstream end of the burner pipe (see Fig. 4.1). Finally the burner pipe exit has been smoothened to prevent any surface roughness or burr from disturbing the straight flow.
The burner pipe diameter $d$ was chosen to be 26 mm out of multiple considerations. First of all, to prevent flashback, the bulk velocity $V_b$ of the DNG/air mixture must be higher than the maximum burning velocity $S$. The maximum laminar burning velocity $S_L$ of a DNG/air mixture is equal to 0.36 m/s. Figure C.1 in the Appendix shows the bulk velocity $V_b$ as a function of the jet Reynolds number defined as $Re = U_b d / ν$ for an inner pipe diameter $d = 26$ mm. For $Re > 2000$, $S$ will increase with Reynolds number as explained in Section 2.2.4. Another reason the pipe diameter $d$ was chosen to be 26 mm was that the size of the CCD chip (2 cm) is ideally a factor of 2-3 smaller than the field of view to be able to use the available camera lenses. A third and final reason was that the cylinder wall had to have a sufficiently large size for the miniature thermocouples to be build into. The size of the burner pipe scales with the size of the cylindrical wall because the ratio of cylindrical outer wall diameter to burner pipe inner diameter $D/d$ was ideally in the vicinity of 3-4 as explained in Section 4.2.
The distance between burner pipe exit and the cylinder wall is denoted by $H$. The ratio $H/d$ can be varied accurately between $H/d = 0$ and $H/d = 7$.

The burner pipe is connected to a similar horizontal pipe. The horizontal pipe is on its turn connected to two similar copper pipes. One of these pipes supplies the DNG and the other supplies the air. To ensure a good homogeneous mixture of DNG and air, a mixer has been inserted into the horizontal pipe. This mixer is a 0.5 m long aluminum strip with a width less than a millimetre smaller than the inner radius of the pipe to ensure a tight fit. The strip has been cut and twisted randomly to force the flow in various directions, mixing the DNG and air along its path. A small part of this mixer is shown in Fig. C.2 in the Appendix.

The two smaller copper pipes supplying DNG and air are connected to flow meters and valves through reinforced plastic tubing. Between the pipe supplying the air and the air flow meter, a cyclone seeder is placed. The seeder is considered part of the PIV configuration and will be discussed in detail in Chapter 4.4. The valves and flow meters are used to set the flow speed of the DNG and air to the desired values. The flow meter for the DNG is a Fisher Porter $FP-1/2-17-G-10/60$ with a $1/2$-GUSVT-410 floater. The flow meter for the air is a Fisher Porter $FP-1/2-27-G-10/60$ with a $1/2$-GUSVT-48 floater. Both flow meters were “calibrated” against a Bronkhorst $F-203AV-M50$ high accuracy Mass Flow Controller (MFC). The inaccuracy of the MFC is much smaller than the reading error for the flow meters and is therefore neglected. The “calibration” results are shown in Figs. C.3 and C.4 in the Appendix.

### 4.1.2 Flame Power

The heating value of DNG is approximately 32 MJ/m$^3$. Under stoichiometric combustion, the DNG/air ratio equals $1/8.5$. The kinematic viscosity $\nu$ of both air and DNG at room temperature is equal to $14.7 \times 10^{-6}$ m$^2$/s. The flame power as a function of the Reynolds number is therefore $P_{\text{flame}} = 1.09 \times \text{Re W}$.

### 4.2 Cylinder Wall

#### 4.2.1 Cylinder Geometry

The cylinder wall as shown in the schematic representation of the experimental setup in Fig. 4.1 and in the photograph taken in Fig. 4.2 is a 0.8 m long cylinder with an outer diameter $D$ of 88.9 mm and a wall thickness $t$ of 3.2 mm.

The cylinder is made of heat resistant Nickel-chromium stainless steel (SAE type 314). The main reason for the curved surface is to reduce the difficult optical access in the near wall region for PIV measurements and reduce spurious scattering. These arguments
would have also applied to a convex plate, however the non-axisymmetric geometry of the cylinder is much more challenging and has therefore been studied much less, creating a gap in knowledge. The cylinder diameter to burner pipe diameter ratio $D/d$ has a value of 3.42 and is a fixed quantity in this experiment. This was partly based on an experiment done by Cornaro and Goldstein (1999) in which they visualized the flow field of a round air jet impinging on cylindrical surfaces with smoke wires. Even though the flames in this experiment would modify the flow field, it still gives a good insight of the behaviour of the flow impinging normally on a cylinder with a similar $D/d$ and Reynolds number. The cylinder has been polished around the stagnation point to ensure a smooth wall. After this the wall has been spray painted black to reduce surface reflections.

![Fig. 4.2: Photograph of the cylinder wall before it was spray painted black. The burner pipe is shown on the bottom. On the top, a pipe is shown entering the cylinder to cool it from within. The polished surface around the area of impingement clearly shows.](image)

### 4.2.2 Thermocouples

The temperature at several positions at the inside of the cylinder is measured by ten J-type miniature thermocouples. J-type thermocouples were chosen from Table 3.2 for their ideal temperature range, high sensitivity, high Curie point, relatively low price and availability. A single thermocouple as shown in Fig. C.5 in the Appendix, has a coat diameter of 0.5 mm and is joined together at the ends. The thermocouples have a length of 1.0 m and can resist temperatures up to 750 °C. They are connected to a *OM-DAQ-USB-2400* Data AcQuisition (DAQ) module by compensation cables. The DAQ module has 8 differential inputs and provides an accuracy of $\pm 1.1$ °C. The thermocouples are positioned carefully inside the cylinder as shown in Fig. 4.3 at the positions according to Figs. C.6 and C.7 in the Appendix. The thermocouples have been positioned such that most thermocouples are around the impingement region: One thermocouple at the stagnation point and one at a distance equal to both half the burner pipe radius and the burner pipe radius in both radial and longitudinal direction.
The other thermocouples have been placed at positions further away from the stagnation point to be able to measure over a large range. Figure C.8 in the Appendix shows the numbers that have been assigned to the thermocouples that will be used to refer to them in the experiments. The thermocouples have been point welded to ensure a good connection between thermocouple and cylinder. The thermocouple wires have been placed in such a way that the wires have a spacing in between each other to prevent any local heating as a result of thermocouple wires shielding the cylinder wall from the impinging cooling air. To ensure the thermocouple wires stay in place, small metal strips have been point welded over them.

![Image of thermocouples](image)

**Fig. 4.3:** The ten J-type thermocouples carefully positioned inside the cylinder by point welding them to the cylinder surface. Metal strips keep the wires in place.

### 4.2.3 Temperature Profile

The outer wall temperature $T_{wh}$ is what is of interest while the thermocouples measure the inner wall temperature $T_{wc}$. The inner wall temperature $T_{wc}$ can be set by controlling the normally impinging air jet volumetric flow rate $Q$ supplied by the side channel blower. It is thus necessary to calculate the temperature difference over the cylinder wall $T_{wh} - T_{wc}$ as a function of the air jet volumetric flow rate $Q$ and the flame jet Reynolds number $Re_h$. This is calculated in Appendix A. The results are shown in Figs. 4.4 and 4.5.

Figure 4.4 shows the cylinder inner wall temperature $T_{wc}$ as a function of the air jet volumetric flow rate $Q$ for different flame jet Reynolds numbers $Re_h$. From Figure 4.5 shows the temperature difference $T_{wh} - T_{wc}$ over the cylinder wall. It shows that the temperature difference $T_{wh} - T_{wc}$ is not very large and almost constant for a certain flame jet Reynolds number $Re_h$. Outside the stagnation zone the Nusselt numbers of the impinging air and flame jet are lower and the temperature difference will therefore be smaller.
Fig. 4.4: The cylinder inner wall temperature $T_{wc}$ as a function of the air jet volumetric flow rate $Q$ for different flame jet Reynolds numbers $Re_h$.

Fig. 4.5: The difference between cylinder inner and outer wall temperature $T_{wh} - T_{wc}$ as a function of the air jet volumetric flow rate $Q$ for different flame jet Reynolds numbers $Re_h$. 
4.3 Cooling Configuration

To cool the cylinder outer wall to a significantly lower value than the flame jet temperature $T_h$, the cylinder is cooled by a normally impinging air jet from the inside (see Fig. 4.2). A minimum cylinder outer wall temperature profile with a mean temperature between 300 °C and 500 °C is aimed for. From Fig. 4.4 it can be seen that for $Re_h = 5000$, a maximum air jet volumetric flow rate $Q$ of 140 m$^3$/h is required. The air jet will be produced by a side channel blower. To choose a suitable side channel blower it is required to compare its pressure drop-volumetric flow rate characteristics with those of the cooling configuration. The side channel blower can deliver a certain volumetric flow rate $Q$ at a certain pressure drop $\Delta p$. The pressure drop $\Delta p$ is caused by friction in the pipes through which the air flows and the sudden change of momentum caused for example by corners. The pressure drop-volumetric flow rate characteristics are calculated in Appendix B.

The result is shown in Fig. 4.6. In this figure, the pressure drop-volumetric flow rate characteristics of two potential Dutair Side Channel Blower candidates (DB307 and DB411) also have been plotted. It shows that for the DB307 and DB411, a maximum volumetric flow rate of 173 and 231 m$^3$/h, respectively can be acquired. The requirement of a flow rate $Q$ of 140 m$^3$/h shows that the DB307 will suffice.

![Fig. 4.6](image_url)

Fig. 4.6: The pressure drop $\Delta p$ of the cooling configuration as a function of the volumetric flow rate $Q$ of the side channel blower. The pressure drop-volumetric flow rate characteristics of the DB307 and DB411 side channel blowers have also been plotted in the same figure to determine the maximum volumetric flow rate that can be acquired.
4.4 PIV Setup

4.4.1 Laser

The laser used is a Quantronix Darwin-Duo 527-80-M. It is a dual oscillator/single head, high repetition rate diode-pumped Nd:YLF laser. It has a pulse-width of less than 120 ns, a wavelength $\lambda$ of 527 nm and a pulse energy of 50 mJ. Repetition rates of up to 10 kHz can be reached. Because it uses two identical oscillators, the highest possible degree of cross-correlations can be reached.

4.4.2 Laser Optics

The formation of a thin planar laser light sheet is achieved through a series of mirrors and lenses as shown in Fig. 4.7.

Fig. 4.7: The laser optics as used in the experiments. On the right side of the photograph, two mirrors and one spherical lens are shown. On the left side, three cylindrical lenses are closely packed together. The Quantronix Darwin-Duo 527-80-M is also shown.

On the right side of the photograph, two mirrors and one lens are shown. The mirrors have the function to redirect the laser light, while the lens converts the slightly diverging laser bundle into a parallel laser bundle. This lens is a double convex spherical lens with focal length $f = +1000$ mm positioned at 400 mm from the laser head. On the left side of the photograph (somewhat hard to see) three lenses are positioned. These three lenses form the thin planar laser light discussed in Section 3.1.2. The first cylindrical lens is a plano-concave cylindrical lens with focal length $f = -38.1$ mm, coated with anti-reflective coating for the visible wavelengths, N-BK7 refractive index and positioned at
1420 mm from the spherical lens. The second and middle cylindrical lens is a plano-convex cylindrical lens with focal length $f = +250$ mm, coated with anti-reflective coating for the visible wavelengths, N-BK-7 refractive index and positioned at 150 mm from the first cylindrical lens. The last cylindrical lens is a plano-convex cylindrical lens with focal length $f = +500$ mm, coated with anti-reflective coating at the visible wavelengths and positioned at 215 mm from the previous cylindrical lens and at 250 mm from the burner pipe centreline.

### 4.4.3 Tracer Particles

The tracer particles used are aluminum oxide particles with the chemical formula $\text{Al}_2\text{O}_3$. They have a diameter $d_p$ of roughly 1 $\mu$m. The particles are heat resistant, follow the flow precisely and do not alter the flow.

A cyclone seeder is inserted as a seeding mechanism in order to supply the flow with the desired tracer particle density. A cyclone seeder is a cylinder that contains a small amount of seeding and has an inlet at the bottom and an outlet at the top. Due to the cylindrical shape and the inlet at the bottom, the air swirls to the top, mixing seeding along its way. A bypass has been inserted with a valve to regulate the amount of seeding. Because generally, a higher seeding density results in better cross correlations and a too high seeding density is very hard to achieve, the bypass was fully closed throughout the experiment.

### 4.4.4 Camera

The camera used to capture the tracer particles is a *Fastcam SA1.1 Photron* high-speed camera. It has a maximum frame rate of 5400 fps at its maximum resolution of 1024 $\times$ 1024 pixels. The sensor is a 12-bit ADC with 20 $\mu$m pixels. It has 8 GB of build in memory which allows it to store 5457 frames at its maximum frequency. The camera is mounted on a rail for easy positioning and focusing.

Two different lenses are used to be able to vary the field of view. The first lens is a Nikon *AF DC-NIKKOR 135 mm f/2 D* with a Tiffen 72 mm (+1) close-up lens to lessen the minimum focal length. The field of view of is 2.9$d \times 2.9d$. The magnification $M$ is therefore equal to 0.27. The $f^\#$ of the lens is set to 4. Using Eqs. 3.3 and 3.4 the particle image diameter $d_e$ can be calculated to be 6.5 $\mu$m, equal to approximately 0.32 pixels. This is too small (see Section 3.1.3) for good cross-correlations, therefore the image is slightly defocused to enlarge the images of the tracer particles.

The second lens is a Nikon *AF MICRO NIKKOR 200 mm f/4 D*. The field of view of 1.8$d \times 1.8d$. The magnification $M$ is equal to 0.44. The $f^\#$ of the lens is set to 5.6, resulting in a particle image diameter $d_e$ equal to 10 $\mu$m, or 0.5 pixels. This is still a bit on the small side but after a slight defocus, PIV analysis shows it suffices.
4.5 Safety Precautions

4.5.1 Ultra Violet Sensor

A ultraviolet sensor has been placed near the flame jet. By pressing a button, the DNG starts to flow and the flame jet can be ignited. As long as the ultraviolet sensor gives off a signal, the DNG keeps flowing. When the flame jet is extinguished as a result of a blow-off, flashback or something else, the ultraviolet sensor gives off no signal and the DNG inlet is shut by a magnetic valve. This precaution makes sure a flashback cannot do much damage or prevents any DNG from flowing into the room when the flame is extinguished. It takes a couple of minutes before the magnetic valve opens up again. At this point the button has to be pressed again to start the DNG flow and ignite the flame jet.

4.5.2 Shielding

The PIV laser is positioned at eye height. To protect the eyes, shielding has been built around the laser light. This shielding consists of metal plates that have been painted black to reflect as little laser light as possible. As an extra precaution, safety goggles are worn that filter out the laser light.
Chapter 5

Results

Section 5.1 focuses on the visual appearance of the stable flames that are produced for different combinations of equivalence ratio and jet Reynolds number. In Section 5.2, mean velocities and Reynolds stresses in the impinging jet region and the wall jet region for one selected flame are investigated. In Section 5.3, this flame is investigated further by looking at the effect of the wall temperature on the mean flow, the Reynolds stresses, the thickness of the unburned layer and the vortex birth location.

5.1 Flame Appearance

5.1.1 Stability Map

The aim of this experiment is to determine which combinations of equivalence ratio and jet Reynolds number produce stable flames. The jet Reynolds number is defined as $Re = U_b d/\nu$. Generally, flames are unstable when the combination of equivalence ratio and jet Reynolds number results in flashback or blowoff. Flashback is when the flame propagates into the burner pipe due to a burning velocity that is larger than the flow velocity. Blowoff occurs when the burning velocity is (much) smaller than the flow velocity and no steady state position of the flame front can be found.

The procedure to determine a stability map is as follows. For a constant air flow rate, the DNG flow rate is chosen such that the mixture is very rich. The DNG flow rate is then slowly reduced until a certain equivalence ratio is reached, where the flame is no longer stable and blowoff or flashback occurs. This value of the equivalence ratio is called the 'critical equivalence ratio' $\Phi_c$. At this point the air flow is slightly increased and the procedure is repeated. Measurements have been performed for jet Reynolds numbers between 1500 and 5000. The experiment was carried out for values of $H/d$ between 1 and 7 to investigate the effect of the wall distance on stability. It was found that $H/d$ has no effect on the stability. The resulting stability map is shown in Fig. 5.1.

Three different regions can be distinguished. For $Re < 2100$ the flow in the pipe is laminar. In the laminar region the critical equivalence ratio $\Phi_c$ decreases sharply with increasing jet Reynolds number. This is caused by an increase in mixture velocity and the relation between the equivalence ratio and the burning velocity as shown in Fig. 2.1.
Below the critical equivalence ratio flashback occurs. For Re > 2100 the laminar pipe flow is no longer stable and velocity fluctuations appear in a gradual transition from laminar to fully turbulent flow. The velocity fluctuations increase the burning velocity, resulting in the rate of decrease of equivalence ratio with jet Reynolds number to be lower than in the laminar region. The stability is still limited by flashback. The relation between the equivalence ratio and the burning velocity is clearly visible when the critical equivalence ratio approaches \( \Phi = 1.0 \). Further decrease of equivalence ratio reduces the burning velocity resulting in blowoff in the region given by Re > 3200. For Re > 3750 the critical equivalence ratio \( \Phi_c \) is a constant of approximately 0.85. This shows that the burning velocity \( S_T \) scales linearly with the Reynolds number in this region.

### 5.1.2 Visual Appearance of the Flames

To study the visual appearance of the flames, color photographs of the stable flames were taken with a commercial camera (Sony CyberShot DSC-HX50V). The results are shown in Fig. 5.2. The centre of each image corresponds to the equivalence ratio and jet Reynolds number of that flame. The camera settings (including the exposure time) were the same for each image.
Above the red line the flames are long diffusion flames with relatively low luminosity. The fuel mixture is very rich and can react only after some mixing with the ambient air. Also visible (especially at the lower jet Reynolds numbers) are the flame bulges which are produced by the regular formation of vortices just downstream of the pipe exit. The images below the red line indicate double-flames where a high luminosity inner flame is surrounded by a (low luminosity) diffusion flame. The length of the inner flame decreases with decreasing equivalence ratio because the burning velocity increases and a steady state position of the inner flame is found at a position closer to the burner pipe exit (where the flow speed is higher). The length of the inner flame stays constant for $Re > 3750$ confirming the linear scaling of $S_T$ with the Reynolds number as discussed in Section 5.1. The luminous intensity of the inner flame increases with decreasing equivalence ratio because the flame area decreases and the amount of fuel reacting in the inner flame increases. The length of the diffusion flame decreases with decreasing equivalence ratio because more fuel reacts in the inner flame and thus less fuel is left for the diffusion flame. Flame $ff$ to $ii$ are single-flames consisting of a inner flame with no surrounding diffusion flame because all fuel reacts in the inner flame. In flames $o$ and $v$ the inner flame is steady and laminar.
5.2 Mean Flow and Reynolds Stresses

To study flame-wall interaction in some detail, attention was focused on only one flame in the stability map. Flame $x$ with $\Phi = 1.3$ and $Re = 3250$ was (somewhat arbitrarily) selected for this purpose. The cylinder was positioned at a distance of $2d$ from the pipe exit, i.e., $H/d = 2.0$. A double-flame with a long inner flame (with cold unreacted gases inside) is surrounded by a diffusion flame (with hot combustion gases and fuel). Both curve around the cylinder wall. A color photograph taken with the earlier mentioned commercial camera of this particular flame and pipe-wall distance is shown in Fig. 5.3.

![Color photograph of flame $x$ with $\Phi = 1.3$, $Re = 3250$ and $H/d = 2.0$.](image)

The cylinder has a stationary wall temperature distribution corresponding to maximum air cooling capacity. Thermocouple readings and outer wall temperatures are shown in Fig. 5.4. The outer wall temperatures are calculated by adding 15 °C to the thermocouple reading as discussed in Section 4.2.3 and illustrated in Fig. 4.5. The temperature distribution shows that the wall temperature has a local maximum at a radial position of 30 mm, where the inner flame touches the wall. Due to the difference between the assumptions made in Section 4.2.3 and what is actually the case, the temperature must be somewhere in between the thermocouple reading and the outer wall temperature given in Fig. 5.4. The uncertainty of thermocouple measurements is negligible.

![Temperature at different radial positions. The solid spheres represent the thermocouple readings. The asterisks represent the outer wall mean temperature by adding 15 °C to the thermocouple reading as discussed in Section 4.2.3.](image)
The mean flow properties were investigated with PIV for a large field of view (2.9d × 2.9d) focused on the impinging flame jet and a smaller field of view (1.8d × 1.8d) focused on the wall jet. The Reynolds stresses are reported only for the smaller field of view. 5 PIV data sets each with 1000 double frames were acquired to determine mean velocities and Reynolds stresses. The repetition rate of the double frames was set to 1500 Hz so that the total measurement time is 3.3 s.

5.2.1 Impinging Flame Jet Region

The first camera lens as described in Section 4.4.4 was used in order to obtain the large field of view (2.9d × 2.9d). A single snapshot of the seeding particle images is shown in Fig. 5.5

![Figure 5.5: A snapshot of the seeding particles for the large field of view. (1) Unreacted cold core. (2) Wall jet region. (3) Large scale vortex. (4) Inner flame. (5) Hot diffusion flame.](image)

The figure shows the unreacted cold core (1) enclosed by the inner flame (4) (and the wall). The inner flame is surrounded by a hot diffusion flame (5) with combustion gases and fuel. The fuel-air mixture exiting the pipe is sufficiently rich for the unreacted cold core to reach the wall and bend around it. At this point a wall jet (2) is formed that also consists of a cold unreacted layer, a inner flame and a hot diffusion flame. Further downstream only a diffusion flame is present because all air from the fuel-air mixture has reacted. Large scale vortices (3) occur periodically as a result of the shear layer between the diffusion flame and the stagnant ambient air.

Velocity vectors were calculated with PIV for interrogation areas with size 32 × 32 pixels. The pulse delay time Δt was set to 150μs. Laser cavity A and B were set to 60
and 64 % of their maximum intensity to minimize wall reflections while light scattered by the tracer particles was sufficient to obtain good quality images. Figure 5.6 shows the mean velocity field determined from 5000 instantaneous velocity fields.

**Fig. 5.6:** The mean velocity field for a set of 5000 images and a measurement time of 3.3 s. The reference vector in the top left corner has a length of $2V_b$. The mean position of the inner flame (as observed in Fig. 5.5) is shown by the dashed lines.

To investigate the mean velocity field in the impinging jet region in detail, the x, and y-components of the mean velocity at various $y/d$ stations located between the burner pipe exit at $y/d = -2.0$ and the cylinder wall at $y/d = 0$, are shown in Fig. 5.7. The velocity components have been made dimensionless with the bulk velocity $V_b$.

The mean transverse velocity component $v_x$ is zero at the jet centreline for all $y/d$ stations indicating good mean flow symmetry. Some distance downstream of the pipe exit the profile for $v_x$ develops a local maximum around $x/d = 0.5$. This coincides with the location of the bright luminous inner flame visible in Fig. 5.3. The combustion increases the velocity component in the x-direction thus tilting the velocity vector outwards. Further downstream the mean transverse velocity component $v_x$ also becomes significant for $x/d > 0.5$. This is caused by the bending of the wall normal flow to form a wall jet.

The streamwise mean velocity $v_y$ at $y/d = -1.90$ gives a good indication of the mean velocity profile at the pipe exit. Further downstream the jet spreads while the centreline velocity remains more or less constant up to $y/d = -0.5$. The centreline velocity $U_c$ is equal to $1.43V_b$. This is consistent with the result of Goldstein (1969) who investigated the effect of jet Reynolds number on the value of $U_c/V_b$, and found $U_c/V_b = 1.42 \pm 0.04$ for $Re = 3250$. The investigation shows that this Reynolds number is at the end of the
transitional region close to the fully turbulent region. For stations $y/d > -0.50$, there is a small dip in mean streamwise velocity around $x/d = 0.6$ caused by the expansion due to combustion at the inner flame.

![Graphs showing mean transverse and streamwise velocity components](image)

**Fig. 5.7:** The mean transverse velocity component $v_x$ and the mean streamwise velocity component $v_y$ measured at various $y/d$ stations located between the burner pipe exit at $y/d = -2.0$ and the cylinder wall at $y/d = 0$. 
5.2.2 Wall Jet Region

PIV measurements were performed for the smaller field of view of $1.8d \times 1.8d$ (see Section 4.4.4) to study the wall jet region. A single snapshot of the seeding particle images is shown in Fig. 5.8

![Fig. 5.8: A snapshot of the seeding particles for the small field of view. (1) Unreacted cold core. (2) Unburned layer. (3) Diffusion flame. (4) Large scale vortex. (5) Inner flame. The impinging jet region and wall jet region are roughly illustrated by the red and yellow borders, respectively.]

The figure shows the unreacted cold core (1) enclosed by the inner flame (5). The inner flame is surrounded by a hot diffusion flame (3) with combustion gases and fuel. In the wall jet region, a cold unreacted layer is visible that will be referred to as the ‘unburned layer’ (2). The thickness and length of this layer will vary in time and this causes large fluctuations in local velocity and temperature. This will be studied in detail later in Subsection 5.3.1. The inner flame (5) exhibits an irregular wavy motion due to a velocity difference between the unreacted cold core and the diffusion flame. The image also shows a large scale vortex (4). These vortices appear occasionally in the field of view. The borders of the impinging jet region and a wall jet region are roughly illustrated by red and yellow lines, respectively.

Velocity vectors were determined with PIV for interrogation areas with size $32 \times 32$ pixels. The pulse delay time $\Delta t$ was set to $150\mu s$. Laser cavity A and B were set to 60 and 64 % of their maximum intensity. Figure 5.9 shows the mean velocity field determined from 5000 instantaneous velocity fields. The dashed line represents the mean location of the inner flame.

The wall-normal jet flow decelerates significantly near the impingement wall. The flow
deflects into a tangential flow that is accelerated by the pressure decrease away from the stagnation point. The maximum mean velocity (marked by a sphere in Fig. 5.9) occurs in the wall jet region and has a value of $1.94 V_b$.

Fig. 5.9: The mean velocity field determined from 5000 images and a measurement time of 3.3 s. The reference vector in the top left corner has a length of $2V_b$. The mean position of the inner flame is illustrated by the dashed line.

Fig. 5.10: An instantaneous velocity field. The reference vector in the top left corner has a length of $2V_b$. The mean position of the inner flame is illustrated by the dashed line. The maximum velocity is illustrated by a sphere.
Figure 5.10 shows an instantaneous velocity field. The velocity vectors of the hot diffusion flame in the impinging jet region are tilted outwards due to combustion of the inner flame. In comparison to the mean velocity field in Fig. 5.9, the instantaneous velocity field is much less smooth, suggesting the presence of Reynolds stresses. Large differences can be seen by visual comparison of the instantaneous and mean velocity fields. In the impinging jet region this is caused by the wavy motion of the inner flame due to a velocity difference between the (faster) cold unreacted core and the (slower) hot diffusion flame. In the wall jet region the difference between instantaneous and mean velocity field is caused by variation in thickness and length of the unburned layer. In the top right part of the vector fields the large difference is caused by the occasional large scale vortex that pulls gases away from the wall into the vortex.

To further investigate the mean flow and Reynolds stresses in the wall jet region, the mean velocities $v_r, v_t$ and the Reynolds stresses $v'_r v'_r, v'_t v'_t$ and $v'_t v'_r$ are determined along radial stations for several values of the angle $\alpha$ between $\alpha = 0^\circ$ and $60^\circ$ using the cylindrical coordinate system as given in Fig. 5.11. The origin is defined at the centre of the cylinder. The radial stations start at $r/(0.5D) = 1.0$ and point radially outward up to $r/(0.5D) = 1.3$. The position of these radial stations depicted as yellow lines.

![Fig. 5.11: Radial stations (yellow lines) where the mean velocity components $v_r, v_t$ and the Reynolds stresses $v'_r v'_r, v'_t v'_t$ and $v'_t v'_r$ were determined. The radial stations are located at angles $\alpha$ of $0^\circ, 6^\circ, 12^\circ, 20^\circ, 30^\circ, 40^\circ, 50^\circ, 55^\circ$ and $60^\circ$ (green lines). The radial stations start at the cylinder wall (orange line) at $r/(0.5D) = 1.0$ and end at $r/(0.5D) = 1.3$ (blue line).](image)

The PIV software determined the velocity components and Reynolds stresses in the
With increasing angle \( \alpha \), the mean radial velocity component \( \overline{v_r} \) decreases. At \( \alpha = 0^\circ \) the value of \( \overline{v_r}/V_b \) at \( r/(0.5D) = 1.29 \) is found to be -1.30 which shows good agreement with the value of \( \overline{v_r}/V_b \) at \( y/d = -0.50 \) in Fig. 5.7, found to be 1.29. For all angles, \( \overline{v_r} \) is zero at the cylinder wall.

A wall jet profile is visible between \( \alpha = 20^\circ \) and 60\(^\circ\). From \( \alpha = 20^\circ \) up to 55\(^\circ\) the wall jet profile increases in thickness and in its maximum value. The maximum mean tangential velocity component \( \overline{v_t} \) has a value of 1.94\(V_b\) and is reached at \( \alpha = 55^\circ \). The maximum of the mean tangential velocity component \( \overline{v_t} \) moves away from the wall as the angle increases. The mean tangential velocity component \( \overline{v_t} \) at \( r/(0.5D) = 1.0 \) is not zero as it should be according to the boundary conditions. This is caused by the fact that the velocity gradient near the wall is very high so that it cannot be resolved by the finite size of the interrogation areas.

The results for the Reynolds stresses \( \overline{v'_r v'_r}, \overline{v'_t v'_t}, \) and \( \overline{v'_r v'_t} \) are shown in Figs. 5.13 and 5.14. The Reynolds stresses were made dimensionless with \( V_b^2 \).

The Reynolds stresses show that there is little to none difference between the mean flow field and instantaneous flow fields at \( \alpha = 0^\circ \) and \( \alpha = 6^\circ \). At \( \alpha = 12^\circ \) the Reynolds stresses \( \overline{v'_r v'_r} \) and \( \overline{v'_t v'_t} \) show a local maximum at \( r/(0.5D) = 1.2 \). These Reynolds stresses are caused by the irregular wavy motion of the inner flame resulting from the velocity difference between the (fast) unreacted cold core and the (slow) diffusion flame. The turbulence kinetic energy produced here will be transported downstream and slowly dissolves as shown by the decreasing amplitude of the local maximum at \( \alpha = 20^\circ \), 30\(^\circ\) and 40\(^\circ\). A small part of the Reynolds stresses present at these angles are also caused by the changing length and thickness of the unburned layer. The Reynolds stresses \( \overline{v'_r v'_r}, \overline{v'_t v'_t}, \) and \( \overline{v'_r v'_t} \) at \( \alpha = 40^\circ \) up to 60\(^\circ\) are caused by large scale vortices. Figures 5.9 and 5.10 show the difference between the mean velocity field and an instantaneous velocity field with a vortex. The vortex increases the tangential velocity at positions further away from the wall \( r/(0.5D) = 1.2 \), while it decreases the tangential velocity at positions near the wall \( r/(0.5D) = 1.1 \).
Fig. 5.12: The mean radial velocity component $v_r$ and the mean tangential velocity component $v_t$ measured at multiple radial stations.
Fig. 5.13: The Reynolds stresses $\overline{v_r v_r}$ and $\overline{v_t v_t}$ measured at multiple radial stations.
Fig. 5.14: The Reynolds stress $\overline{v_t v_r}$ measured at multiple radial stations.
The turbulence kinetic energy \( k \) is illustrated as a surface plot in Fig. 5.15. The production of turbulence kinetic energy at a small angle (\( \alpha = 12^\circ \)) and at larger angles (\( \alpha = 40^\circ \) up to \( 60^\circ \)) is visible. To recap, at \( \alpha = 12^\circ \) this is caused by the wavy motion of the inner flame caused by the velocity difference between the (faster) cold unreacted core and the (slower) hot diffusion flame. At higher angles the turbulence kinetic energy is caused by the large scale vortices that decrease the tangential velocity near the wall and increase the tangential velocity at the outer edge of the diffusion flame.

**Fig. 5.15:** The turbulence kinetic energy \( k \) illustrated by a surface plot. The black full line represents the cylinder wall. The dashed yellow lines represent the angles of the radial stations. The dashed black lines represent the jet centreline and burner pipe inner radius.

### 5.3 Effect of Wall Temperature

This section reports on the effect of the wall temperature. Two stationary wall temperature profiles will be considered. A low temperature distribution and a high temperature distribution that correspond to maximum air cooling capacity and no cooling, respectively. Thermocouple readings and outer wall temperatures are shown in Fig. 5.16 for both the cold and hot wall cases. Both temperature distributions reach a maximum at a circumferential distance of 30 mm (\( \alpha \approx 39^\circ \)) because the inner flame touches the wall around this circumferential distance as shown in Fig. 5.11.

#### 5.3.1 Unburned Layer Thickness

The aim of this investigation is to determine the effect of the wall temperature on the thickness of the unburned layer \( \delta \). The unburned layer thickness \( \delta \) is defined as the distance between the cylinder wall and the inner flame. The unburned layer thickness \( \delta \) was measured at radial stations for several values of the angle \( \alpha \) between \( 12^\circ \) and \( 22^\circ \) in steps of \( 2^\circ \). The difference in wall temperature between the cold and the hot wall is approximately \( 400 \, ^\circ \mathrm{C} \) as shown in Fig 5.16.
The laser and camera were set to a frequency of 1500 Hz. Only laser A was used and was set to 60% of its maximum intensity. For both the cold and the hot wall cases, a total of 18000 raw images of seeding particles were obtained and processed with a Matlab program to extract the unburned layer thickness $\delta$ for each angle $\alpha$. The result of the Matlab program on a single raw image of seeding particles is shown in Fig. 5.17.

**Fig. 5.16:** Temperature as a function of the circumferential distance. The solid spheres represent the thermocouple readings. The asterisks represent the outer wall mean temperature by adding 15°C to the thermocouple reading as discussed in Section 4.2.3. The green dashed lines represent the angles $\alpha$ of the radial stations used in the experiments.

**Fig. 5.17:** A raw image of seeding particles with image processing results plotted on top. The green line represents the border between regions of high and low density. The blue striped line represents the cylinder wall. The yellow striped lines represents the angles at which the thickness is measured. The red solid lines represent the thickness of the unburned layer $\delta$. The light blue striped line represents the maximum angle the unburned layer reaches.
The Matlab program works on the raw images of seeding particles. The basics of the program are as follows. It first removes the background and wall reflections. It then does 250 disk convolutions with a 3 pixel radius to smooth out the high (seeding particle) density unreacted cold core and the low (seeding particle) density diffusion flame, while maintaining the difference in density between these two regions. Binary imaging followed by boundary tracing is then applied to obtain the boundary of the high (seeding particle) density region. The unburned layer thickness δ, which is the distance from the cylinder wall to the inner flame (represented by the boundary) is then stored for each angle α and Fig. 5.17 is produced. A more detailed description is as follows.

<table>
<thead>
<tr>
<th>Step #</th>
<th>Procedure Description</th>
<th>Figure</th>
</tr>
</thead>
<tbody>
<tr>
<td>0: Raw image</td>
<td>Raw image obtained by the camera through Davis.</td>
<td><img src="image1.jpg" alt="Figure" /></td>
</tr>
<tr>
<td>1: Subtraction &amp; Remove wall reflection</td>
<td>A value is subtracted to remove the captured luminescence. Then the wall reflection is removed so that this very high intensity region does not become problematic in the convolution step.</td>
<td><img src="image2.jpg" alt="Figure" /></td>
</tr>
<tr>
<td>2: Scale intensity</td>
<td>The intensity is scaled so that the sum of all the pixels equals 14462626. This is to generalize the upcoming steps. This value looks random but it is the sum of intensities of the first images on which the program was based.</td>
<td><img src="image3.jpg" alt="Figure" /></td>
</tr>
<tr>
<td>Step #</td>
<td>Procedure Description</td>
<td>Figure</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------</td>
<td>--------</td>
</tr>
<tr>
<td>3: Background subtraction</td>
<td>Subtract a pixel value of 50 to eliminate the background and get a better contrast. This helps in the next step.</td>
<td><img src="image1" alt="Image" /></td>
</tr>
<tr>
<td>4: 250 disk convolutions</td>
<td>Apply a convolution with a disk with a radius of 3 pixels. This is done 250 times to smoothen out the high density cold core (and unburned layer).</td>
<td><img src="image2" alt="Image" /></td>
</tr>
<tr>
<td>5: Subtraction</td>
<td>Subtract a pixel value of 10 so the lower density areas have no more intensity. This value was found by trial and error.</td>
<td><img src="image3" alt="Image" /></td>
</tr>
<tr>
<td>6: Binary imaging</td>
<td>Apply binary imaging with a 0.0001 threshold. All pixels in the lower density area’s obtain value 0 and all pixels in the higher density area’s obtain value 1.</td>
<td><img src="image4" alt="Image" /></td>
</tr>
<tr>
<td>Step #</td>
<td>Procedure Description</td>
<td>Figure</td>
</tr>
<tr>
<td>---------</td>
<td>----------------------------------------------------------------------------------------</td>
<td>--------</td>
</tr>
<tr>
<td>7: Filling holes</td>
<td>If there are any pixels with value 0 fully enclosed in pixels with value 1, these pixels with value 0 are 'filled' and set to a value of 1. This happens rarely.</td>
<td><img src="https://example.com/image1.png" alt="Image" /></td>
</tr>
<tr>
<td>8: Boundary tracing</td>
<td>Trace the boundary of all the pixels with value 1, which is the high density region. This boundary is stored and plotted on top of the image from step 1 as a green line.</td>
<td><img src="https://example.com/image2.png" alt="Image" /></td>
</tr>
<tr>
<td>9: Plotting</td>
<td>The cylinder wall is plotted as a dark blue striped line. The six radial lines are plotted as yellow striped lines. Along these lines, the thickness of the unburned layer is measured. A light blue striped line is plotted which represents the length of the unburned layer around the cylinder wall.</td>
<td><img src="https://example.com/image3.png" alt="Image" /></td>
</tr>
<tr>
<td>10: Calculating thickness</td>
<td>The thickness of the unburned layer $\delta$ is calculated for each angle and plotted as a red line with red dots at the boundary intersection. The thicknesses are stored.</td>
<td><img src="https://example.com/image4.png" alt="Image" /></td>
</tr>
</tbody>
</table>
Initially it was also the intention to investigate the maximum angle of the unburned layer. Unlike the thickness, it is very problematic to extract the maximum angle accurately because the front of the layer is often very thin and can disappear in the wall reflections. It is therefore not reported.

The Matlab routine had computed the thickness for all angles for both the cold and the hot wall for all 18000 images. The result is shown in Fig. 5.18 by applying smoothing distribution fits to the histograms of the unburned layer thicknesses \( \delta \). Each line in the graph represents the probability density function of the unburned layer thickness \( \delta \). Dashed lines correspond to the cold wall case and full lines to the hot wall case.

![Fig. 5.18: Probability density function of the thicknesses of the unburned layer \( \delta \) for \( \alpha = 12^\circ \), 14°, 16°, 18°, 20° and 22° for both the cold wall (dashed lines) and the hot wall (full lines).](image)

For the smallest angle (\( \alpha = 12^\circ \)), there is no difference between the cold and the hot wall cases suggesting that the wall temperature has no effect on the unburned layer thickness \( \delta \) at this position. However, for angles between \( \alpha = 14^\circ \) and \( \alpha = 22^\circ \) the effect of the wall temperature is clearly visible and it increases with increasing angle \( \alpha \).

In the hot wall case the unburned layer receives more heat from the wall than in the cold wall case. Therefore, because of thermal expansion the thickness of the unburned layer \( \delta \) is larger for the hot wall case than for the cold wall case. The thickness of the unburned layer \( \delta \) decreases with increasing angle \( \alpha \) due to combustion in the inner flame. Because the thickness \( \delta \) decreases with increasing angle \( \alpha \), the thermal expansion increases with
increasing angle $\alpha$. The difference between the unburned layer thickness $\delta$ in the cold wall and the hot wall cases $\Delta \delta (= \delta_h - \delta_c)$ therefore increases with increasing angle $\alpha$.

Table 5.1 summarizes the results. In the first column, the angle $\alpha$ is shown. In the second and fourth column the mean thickness $\bar{\delta}$ is shown for the cold wall and the hot wall cases. In the third and fifth column the standard deviation $\sigma$ of the thickness $\delta$ over the full set of 18,000 thicknesses is shown for both cases. In the sixth column, the difference $\Delta \bar{\delta}$ is shown. In the seventh column, the 2-D mean thermal expansion of the hot wall case with respect to the cold wall case is shown ($\delta_h/\delta_c$).

<table>
<thead>
<tr>
<th>$\alpha$ ($^\circ$)</th>
<th>$\bar{\delta}_c$ (mm)</th>
<th>$\sigma_c$ (mm)</th>
<th>$\bar{\delta}_h$ (mm)</th>
<th>$\sigma_h$ (mm)</th>
<th>$\Delta \bar{\delta}$ (mm)</th>
<th>$\delta_h/\delta_c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>12</td>
<td>4.72</td>
<td>0.98</td>
<td>4.72</td>
<td>0.93</td>
<td>0.00</td>
<td>1.00</td>
</tr>
<tr>
<td>14</td>
<td>3.64</td>
<td>0.90</td>
<td>3.71</td>
<td>0.87</td>
<td>0.07</td>
<td>1.02</td>
</tr>
<tr>
<td>16</td>
<td>3.06</td>
<td>0.92</td>
<td>3.25</td>
<td>0.88</td>
<td>0.19</td>
<td>1.06</td>
</tr>
<tr>
<td>18</td>
<td>2.66</td>
<td>0.96</td>
<td>2.96</td>
<td>0.91</td>
<td>0.30</td>
<td>1.11</td>
</tr>
<tr>
<td>20</td>
<td>2.34</td>
<td>1.00</td>
<td>2.76</td>
<td>0.93</td>
<td>0.42</td>
<td>1.18</td>
</tr>
<tr>
<td>22</td>
<td>2.09</td>
<td>1.05</td>
<td>2.58</td>
<td>0.96</td>
<td>0.49</td>
<td>1.23</td>
</tr>
</tbody>
</table>

Table 5.1: Summary of the mean unburned layer thickness $\bar{\delta}$ for both cases, standard deviation $\sigma$ for both cases, difference in mean unburned layer thickness $\Delta \bar{\delta}$ and thermal expansion $\delta_h/\delta_c$ for each angle.

### 5.3.2 Mean Flow in the Impinging Flame Jet Region

This investigation focuses on the effect of the wall temperature on the mean flow in the impinging jet region. The cold wall case has already been considered in Section 5.2.1. The setup and settings for the hot wall case were identical to this except that the wall was not cooled. The measurements for the cold and hot wall have been performed successively to ensure nearly identical environmental conditions. Measurements were performed at the same $y/d$ stations as in Section 5.2.1. Figure 5.19 shows the mean velocity components $\bar{v}_x$ and $\bar{v}_y$ for both the cold and the hot wall. The velocity components have been made dimensionless with the bulk velocity $V_b$.

Both the streamwise and transverse velocities are not affected by the wall temperature between the burner pipe exit ($y/d = -2.0$) and $y/d = -0.75$. At distances closer to the wall there is a small effect that increases with increasing $y/d$. The mean transverse velocity component $\bar{v}_x$ is higher for the hot wall case because the wall expands the low temperature region outward (by symmetry). The mean streamwise component $\bar{v}_y$ is lower for the hot wall case at $x/d < 0.5$ because the expansion is in the negative $y$-direction opposing the flow.
Fig. 5.19: The mean transverse velocity component $v_x$ and mean streamwise velocity component $v_y$ measured at various $y/d$ stations for both a cold and hot wall.
5.3.3 Vortex Formation

Large scale vortices can be observed periodically in the field of view. In Section 5.2.2 it was discussed that large scale vortices are the main cause of the Reynolds stresses present in the wall jet region. The effect of wall temperature on the vortex formation is investigated in this section as a prerequisite of the investigation of the effect of wall temperature on the Reynolds stresses (caused by the large scale vortices) in Section 5.3.4.

For both the cold and the hot wall cases, the position of “vortex birth” is considered. The vortex birth is defined here as the first sign of the formation of a large scale vortex. It can be determined from the shape of the interface between the edge of the hot diffusion flame and the ambient air. The vortex birth is defined as the time when the angle between the two segments shown in Fig. 5.20 at $t = 13.2$ ms becomes less than $90^\circ$.

![Fig. 5.20: Formation of a vortex. The time between images is 13.2 ms. The boundary between the high temperature vortex and the stagnant ambient air is illustrated by full white lines. The definition of vortex birth is shown at $t = 13.2$ s.](image)

The first two images (at $t = 0.0$ and $t = 13.2$ ms) show the first moments of a vortex formation. At these moments, the diffusion flame surrounding the inner flame is relatively thick. At the outer edges of the diffusion flame the fuel has (almost) fully reacted. When the diffusion flame thickness increases (due to one of the passing bulges) the outer edges of the diffusion flame receive less of the heat released at the inner flame and the temperature at the outer edges decreases. Because the temperature decreases, the buoyancy decreases and the thickness increases again. The thickness keeps increasing and buoyancy keeps decreasing. The initial formation of a vortex is caused by the velocity difference between the diffusion flame and the ambient air and is enhanced by the decrease in buoyancy at the edges of the thick diffusion flame. In the next few images ($t = 13.2 - 52.8$ ms) the front of the vortex is at a relative standstill while the length
of the vortex increases. The velocity at the edge of the diffusion flame increases as a result of the circular vortex motion until \((t > 52.8 \text{ ms})\) the front of the vortex that rises slowly (due to buoyancy) is close enough to the high temperature region that it is pulled along. This is shown in the last three images \((t = 66.0 - 92.4 \text{ ms})\). When the vortex has passed, the velocity at the edge of the high temperature region decelerates and the thickness increases. The thickness will increase with time again and will combine with a passing bulge to form a new vortex.

For the cold and hot wall cases, 5000 images of the seeding particles (the same as those in Section 5.3.2) were manually processed by visual inspection to determine the position of each vortex birth. The total measurement time for both the hot and cold wall cases was 3.3 s. During this time 42 vortices formed in both cases. A histogram for both the x- and y-coordinates of the vortex birth position for both cases was made. The results are shown in Fig. 5.21. Each cross represents a single measurement. A large cross represents the mean position of vortex birth.

**Fig. 5.21:** Vortex birth positions for the cold and hot wall cases. In the large image, the small crosses depict the position of vortex birth for each of the 42 vortices. The mean position is shown by the large crosses. Histograms of the x- and y-coordinates of the vortex birth positions are shown along the horizontal and vertical axes for both cases.

In general, the position of vortex birth is located at a further downstream position (and thus larger angle \(\alpha\)) for the hot wall case than in the cold wall case. The birth of a vortex is initiated by the previous vortex. In the hot wall case, the gases in the wall jet region are hotter compared to the cold wall case. The increased buoyancy in the hot wall case therefore feeds less gases into the vortex. In general, the vortex formation will
therefore take longer in the hot wall case and the next vortex forms at a further position downstream compared to the cold wall case.

5.3.4 Mean Flow and Reynolds Stresses in the Wall Jet Region

This section compares the mean flow velocity components and the Reynolds stresses for the cold and hot wall cases. The cold wall case has already been considered in Section 5.2.2. The measurements for the cold and hot wall cases have been performed successively to ensure (nearly) identical conditions. Measurements were performed at the same radial stations as in Section 5.2.2. The wall temperatures distribution for the cold and hot wall cases can be found in Fig. 5.16. Figure 5.22 shows the velocity components $v_r$ and $v_t$ for the cold and hot wall cases. The velocity components were made dimensionless with the bulk velocity $V_b$. For $\alpha = 60^\circ$ at $r/(0.5D) > 1.23$ there are no data points because this is outside the field of view.

The effect of wall temperature on the mean flow velocity components $v_r$ and $v_t$ is very small. The mean radial velocity component $v_r$ shows no significant difference at all radial stations. The mean tangential velocity component $v_t$ at $\alpha = 0^\circ$ and $\alpha = 6^\circ$ shows no difference on the mean velocity components since these radial stations are in the cold unreacted core. The largest difference can be seen at $\alpha = 40^\circ$, which is the position where most vortices are born in the hot wall case (see Fig. 5.21). The vortices increase the mean tangential velocity at the outer edge of the diffusion flame. The difference of the mean tangential flow between the cold and hot wall cases is highest at $\alpha = 40^\circ$, the angle where most vortices are born as shown in Fig. 5.21.

Figures 5.23 and 5.24 show the Reynolds stresses $\overline{v'_rv'_r}$, $\overline{v'_tv'_t}$ and $\overline{v'_rv'_t}$ for both the cold and hot wall. The Reynolds stresses were made dimensionless with $V_b^2$.

No differences in Reynolds stresses occurs between the cold and hot wall cases at $\alpha = 0^\circ$ and $\alpha = 6^\circ$. At $\alpha = 12^\circ$ the Reynolds stresses for the hot wall case are increased (in comparison to the cold wall case) around $r/(0.5D) > 1.15$. The Reynolds stresses at this radial station are mainly caused by the velocity fluctuations caused by the wavy inner flame. The amplitude of the wavy inner flame increases with increasing velocity difference between the cold unreacted core and the diffusion flame. In the cold wall case, more vortices are born at a lower $y$-positions. These vortices increase the flow velocity of the diffusion flame, decreasing the velocity difference between the cold unreacted core and the diffusion flame, thus decreasing the amplitude of the wavy interface and so the Reynolds stresses. At $\alpha = 20^\circ$ and $\alpha = 30^\circ$, the Reynolds stresses are mainly caused by the transport of turbulence kinetic energy produced at $\alpha = 12^\circ$. The Reynolds stresses between $\alpha = 40^\circ$ and $\alpha = 60^\circ$ are caused by the formation of vortices as described in Section 5.2.2. The difference between the Reynolds stresses for the cold and hot wall cases at these angles are caused by the position at which vortices are formed. Section 5.3.3 showed that vortices are generally formed at a larger angle. The increased Reynolds stresses at larger angles support this.
Fig. 5.22: The mean radial velocity component $v_r$ and the mean tangential velocity component $\bar{v}_t$ measured at multiple radial stations for both the cold and hot wall cases.
Fig. 5.23: The Reynolds stresses \( \overline{v'_r v'_r} \) and \( \overline{v'_t v'_t} \) measured at multiple radial stations for both the cold and hot wall cases.
Fig. 5.24: The Reynolds stress $\overline{v_t v_r}$ measured at multiple radial station for both the cold and hot wall cases.
The turbulence kinetic energy $k$ for the cold and hot wall cases are illustrated as surface plots in Fig. 5.25. The difference between the hot and cold wall regions are clearly visible and are consistent with the measurements of the Reynolds stresses at the radial stations. The small peak at the front ($\alpha = 12^\circ$) is caused by the wavy motion of inner flame and is higher for the hot wall case. The slope at the end ($\alpha = 40^\circ$ up to $\alpha = 60^\circ$) is caused by the formation of the vortices and is higher for the hot wall case.

**Fig. 5.25:** The turbulence kinetic energy $k$ illustrated by a surface plot for the cold (top) and hot (bottom) wall cases. The black full line represents the cylinder wall. The dashed yellow lines represent the angles of the radial stations. The dashed black lines represent the jet centreline and inner radius.


CHAPTER 6

Conclusion

In this study a flame jet that impinges normally on a cooled cylinder wall was considered. The effect of wall temperature on flow properties such as the mean velocities, Reynolds stresses and flame structure was investigated in a plane normal to the cylinder axis using PIV.

First, a stability map was produced to determine the combinations of jet Reynolds number Re and equivalence ratio Φ that produce stable flames. It was found that for Re < 3200 the flames were limited by flashback. The flashback limited region is subdivided in a laminar region (Re < 2100) and a transitional region(2100 < Re < 3200). Each region shows different rates of decrease of critical equivalence ratio with Reynolds number caused by the increasing turbulent burning velocity $S_T$ for Re > 2100. The region defined by Re > 3200 (and Re < 5000) is limited by blowoff. For Re > 3750 the critical equivalence ratio $\Phi_c$ is a constant showing that the turbulent burning velocity $S_T$ increases linearly with Reynolds number.

Further investigation focused on a double flame with Re = 3250, $\Phi = 1.3$ and $H/d = 2.0$. It was found that the wall temperature has little effect on the mean flow velocities. Small differences in the mean flow velocities are found near the wall for the impinging jet region caused by thermal expansion of the unreacted cold core. In the wall jet region small differences are found at the outer edge of the diffusion flame caused by the location where large scale vortices are formed. Large scale vortices are generally formed at a position further downstream for a hot wall than for a cold wall. It was found that Reynolds stresses are directly and indirectly caused by the vortices. At the point where the inner flame bends around the wall, Reynolds stresses are caused by the wavy motion of the inner flame caused by the shear between the fast unreacted cold core and the slow diffusion flame. The location where vortices are formed leads to decreased Reynolds stresses for the cold wall case. Large Reynolds stresses are also present in the wall jet region at the outer edges of the diffusion flame. These Reynolds stresses are caused by vortices that accelerate the flow at the outer edges and decelerate the flow near the wall. These Reynolds stresses are also higher for the hot wall case caused by the position at which vortices are formed.

It was found that the unburned layer thickness $\delta$ increases with increasing wall temperature due to thermal expansion. Also the thickness decreases with decreasing angle $\alpha$. Finally it was found that the relative increase in unburned layer thickness $\delta$ increases with angle $\alpha$. 
APPENDIX A

Wall Temperature Profile

The stagnation region of the flame is very important. In this region the Nusselt number is usually the highest. The temperature profile in the wall is therefore calculated at the stagnation point, where the wall is approximated by an insulated flat plate. In this case a simple heat transfer balance can be set up if stationary heat transfer is assumed. In general form, the heat transfer rate per unit area is given by

\[ q'' = \frac{Q_{ht}}{A} = h \Delta T, \]  

(A.1)

where \( Q_{ht} \) is the heat transfer rate, \( A \) the area, \( h \) the heat transfer coefficient and \( \Delta T \) the temperature difference. The heat transfer coefficient \( h \) is given by

\[ h = \frac{N_u k_{th}}{L}, \]  

(A.2)

where \( N_u \) is the Nusselt number, \( k_{th} \) the thermal conductivity of the gases and \( L \) the characteristic length. Figure A.1 shows a graphical representation of the flame jet, the air jet, the wall and the temperature profile.

Fig. A.1: Graphical representation of the flame jet, the air jet, the wall and the temperature profile.
The heat transfer rate per unit area $q_1''$, which is the heat transfer rate per unit area from the relatively hot flame to the relatively cold cylinder outer wall can now be expressed as

$$ q_1'' = \frac{Nu_h k_h}{d} (T_h - T_w), $$  \hspace{1cm} (A.3)

where $Nu_h$ is the Nusselt number of the flame to wall heat transfer describing the heat transfer between impinging flame jet and wall. Because mixing and thus advection increases with increasing Reynolds number, the convective heat transfer, and thus the Nusselt number also increases with the increase of the Reynolds number. Experiments have shown that this increase in Nusselt number at the stagnation point scales with $\sqrt{Re}$ (van der Meer, 1987) approximately. A good approximation for the stagnation point Nusselt number of a impinging flame jet is given by Dong and Leung (2001)

$$ Nu_h = 0.38 \sqrt{Re}. $$  \hspace{1cm} (A.4)

The thermal conductivity of the relatively hot mixture of fresh and burnt gases is denoted by $k_h$ and depends on the temperature of the mixture. By approximating the mixture as air and fitting a well-known table of thermal conductivity versus the temperature, the following relation is derived for the thermal conductivity as a function of temperature

$$ k_h = -1.52 \times 10^{-11} T^3 - 4.86 \times 10^{-8} T^2 + 1.02 \times 10^{-4} T - 0.000393. $$  \hspace{1cm} (A.5)

The temperature of the flame $T_h$, is approximately 2150 K (Singh and Dreizler, 2013). The temperature of the outer cylinder wall is denoted by $T_w$.

The heat transfer rate per unit area $q_2''$ is the heat transfer rate per unit area from the cylinder outer wall to its inner wall, given by

$$ q_2'' = \frac{k_w}{l} (T_w - T_w), $$  \hspace{1cm} (A.6)

where $k_w$ is the thermal conductivity of the steel which has a value of 14.2 W/mK. $T_w$ is the temperature at the inside wall of the cylinder. This is the temperature measured by the thermocouples.

The heat transfer rate per unit area $q_3''$ is the heat transfer rate per unit area from the cylinder inner wall to the impinging air jet, given by

$$ q_3'' = \frac{Nu_c k_c}{d} (T_w - T_c). $$  \hspace{1cm} (A.7)
The stagnation point Nusselt number $Nu_c$ for a impinging air jet as a function of the Reynolds number has been calculated experimentally by (Katti and Prabhu, 2008) and is given by

$$Nu_c = 0.726Re_c^{0.53}. \quad (A.8)$$

The thermal conductivity of the normally impinging air jet $k_c$ can also be calculated using Eq. A.5. $T_c$ is the temperature of the impinging air jet which is approximately 300 K. This is slightly higher than room temperature because the side channel blower heats up the air a little. The heat transfer rate is conserved, so $q'' = q_1'' = q_2'' = q_3''$. By combining Eqs. A.3, A.6 and A.7 the following equation is obtained

$$q'' = \frac{Nu_h k_h}{d} (T_h - T_{wh}) = \frac{k_w}{l} (T_{wh} - T_{wc}) = \frac{Nu_c k_c}{d} (T_{wc} - T_c). \quad (A.9)$$

It is now possible to solve for $T_{wh}$ and $T_{wc}$. Because $k_h$ and $k_c$ are temperature dependent, this cannot be done explicitly. The thermal conductivities $k_h$ and $k_c$ are temperature dependent, ranging from $T_{wh}$ to $T_h$ and from $T_c$ to $T_{wc}$ respectively. To approximate these thermal conductivities, they will be based on “bulk” thermal conductivities calculated from a linear mean temperature in these ranges. Because $k_h$ and $k_c$ depend on $T_{wh}$ and $T_{wc}$ a solution must be found iteratively. The results are shown in Figs. 4.4 and 4.5.
APPENDIX B

Pressure Drop - Volumetric Flow Rate Characteristics

The pressure drop $\Delta p$ over a series of pipes and bends can be calculated by (van den Akker and Mudde, 2008, pp. 230) as

$$\Delta p = \rho_{\text{air}} e_{fr},$$

(B.1)

where $\rho_{\text{air}}$ is the density of air and $e_{fr}$ is the energy dissipation given by

$$e_{fr} = \sum_i \left( 4 f_s \frac{L_s}{D_s^2} < v >^2_s \right)_i + \sum_j \left( K w_s \frac{1}{D_s^2} < v >^2_s \right)_j,$$

(B.2)

where $f_s$ is the Fanning friction factor, $L_s$ the length of a section, $D_s$ the hydraulic diameter of a section, $< v >_s$ the mean flow speed of a section and $K w_s$ the friction coefficient of a section. When the pressure drop $\Delta p$ is expressed in terms of the volumetric flow rate $Q$ in units of m$^3$/h, the following expression is obtained

$$\Delta p = \rho_{\text{air}} \frac{8}{\pi^2} \left( \sum_i \left( 4 f_s \frac{L_s}{D_s^2} \right)_i + \sum_j \left( K w_s \frac{1}{D_s^2} \right)_j \right) \left( \frac{Q}{3600} \right)^2.$$

(B.3)

The first part of this equations represents the energy dissipation in all the straight parts of the pipes caused by friction. The second part represents any bents, contractions, valves and such, that causes a change in momentum, which requires work.

With the use of Fig. 4.1, the pressure drop $\Delta p$ as a function of the volumetric flow rate $Q$ is calculated. The inlet of the side channel blower is connected to a plastic tubing of 3.5 m in length and 5.5 cm in diameter which is called section 1. A valve is connected to this plastic tubing to regulate the inflow of air. The valve has a diameter of 7 cm and has a friction coefficient of 0.2 when it is fully opened (Janssen and Warmoeskerken, 2006, pp. 83). This valve is called section 2. The outlet of the side channel blower is connected to plastic tubing of 3.5 m in length and 3.5 cm in diameter which is called section 3. This plastic tubing is connected to a copper pipe of 1.0 m in length and 2.6 cm in diameter. This copper pipe is called section 4. The copper pipe has a smooth
bent of 90° which is called section 5. The smooth bent has a friction coefficient of 0.013 (Janssen and Warmoeskerken, 2006, pp. 84). The copper pipe enters the cylinder wall from the top through a hole in the cylinder. From the copper pipe, the air impinges normally on the inside of the cylinder in order to cool the cylinder. The air then goes two ways. This T-junction is called section 6. Because the inflow diameter is much different from the outflow diameter of the T-junction, the mean of the two is taken as an approximate “effective” diameter. The large cylinder is called section 7. Because the air coming from the copper pipe is divided over the two cylinder exits, the mean flow in the cylinder is two times lower. This is taken into account by setting an “effective” diameter of $4D_s$. All geometries, friction factors and coefficients are shown in Table B.1 (Janssen and Warmoeskerken, 2006, pp. 82-84).

<table>
<thead>
<tr>
<th>Section</th>
<th>$L_s$ (m)</th>
<th>$D_s$ (m)</th>
<th>$4f_s$</th>
<th>$Kw_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.5</td>
<td>0.055</td>
<td>0.024</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>-</td>
<td>0.07</td>
<td>-</td>
<td>0.2</td>
</tr>
<tr>
<td>3</td>
<td>3.5</td>
<td>0.035</td>
<td>0.030</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>1.0</td>
<td>0.026</td>
<td>0.016</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>-</td>
<td>0.026</td>
<td>-</td>
<td>0.013</td>
</tr>
<tr>
<td>6</td>
<td>-</td>
<td>(0.026+0.0899)/2</td>
<td>-</td>
<td>1.0</td>
</tr>
<tr>
<td>7</td>
<td>1.5</td>
<td>4×0.0889</td>
<td>0.016</td>
<td>-</td>
</tr>
</tbody>
</table>

**Table B.1:** Geometries, friction factors and friction coefficients for the different parts of the cooling configuration.

By substituting these values into Eqs. B.3 and B.1 consequently, the pressure drop $\Delta p$ as a function of the volumetric flow $Q$ rate can be obtained. The result is shown in Fig. 4.6.
Appendix C

Figures

Fig. C.1: The bulk velocity as a function of Reynolds number. The dashed line represents the bulk velocity. The full line represents the DNG/air maximum laminar burning velocity.
Fig. C.2: Photograph of a part of the self-made aluminum mixer.
Fig. C.3: The volumetric flow rate of air for the FP-1/2-27-G-10/60 with a 1/2-GUSVT-48 floater as a function of floater position.

Fig. C.4: The volumetric flow rate of DNG for the FP-1/2-17-G-10/60 with a 1/2-GUSVT-410 floater as a function of floater position.
Fig. C.5: A single J-type thermocouple that is used to measure temperature. The 2 euro coin has been included as a size reference.
Fig. C.6: Circumferential positions of six thermocouples placed inside the cylinder. This is the sideview of the cylinder. Positions are given as outer diameter distances in mm.
Fig. C.7: Longitudinal positions of five thermocouples placed inside the cylinder. This is the topview of the cylinder. Longitudinal positions are given in mm.

Fig. C.8: Topview of the cylinder showing all thermocouple positions and the numbers assigned to them.
Bibliography


