Doctoral Thesis

Origin and control of thermoacoustic instabilities in lean premixed gas turbine combustion

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Origin and Control of
Thermoacoustic Instabilities in Lean Premixed
Gas Turbine Combustion

A dissertation submitted to the
SWISS FEDERAL INSTITUTE OF TECHNOLOGY ZURICH
for the degree of
DOCTOR OF TECHNICAL SCIENCES

presented by

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Dr. M. Füri, co-examiner

2005
Acknowledgment

This work was carried out during my time as a research assistant at the Aerothermochemistry and Combustion Systems Laboratory (LAV) at the Institute of Energy Technology, ETH Zurich. At the end of this project, I will take the advantage of the situation to thank all the people who worked, encouraged and supported me during the time when thermoacoustic instabilities were dominating my life!

Specially, I would like to thank Prof. Dr. Konstantinos Boulouchos for the opportunity he gave me to work in his research group, for our fruitful discussions, his continual suggestions and his personal encouragement for this project.

Also, Prof. Dr. Christian Oliver Paschereit has to be thanked for refereeing this thesis. His constant interest, his valuable comments and encouragement for this project will not have been taken for granted.

Of course, a big thank you has to go to Dr. Marc Füri. Without all the stimulating time in the office or down in the laboratory, this work would not have been possible - the Swiss “Röschti”-moat has become a minor matter.

Peter Eberli, Marcel Décosterd and Gerhard Egli are the people who made experimental work for me a little bit easier to handle. Thank you for supporting me.

The entire work was supported by Alstom Power (Switzerland) in the frame of the Center for Energy Conversion (CEC) at the ETHZ. Profitable discussions with Peter Flohr, Bruno Schuermans and Valter Bellucci on thermoacoustic instability mechanisms in swirl-stabilized burners are gratefully acknowledged.

There are many people who have helped me along the way that I did not mention and I hope that I can repay them with the same passionate interest in their problems as they helped me with mine.
Seite Leer / Blank leaf
Abstract

Modern gas turbines use lean premixed combustion to achieve the best compromise between pollutant emissions and efficiency. This type of combustion increases the flame receptivity to external perturbations thereby promoting the onset of large amplitude pressure oscillations called thermoacoustic instabilities. This phenomenon results from the resonant coupling of fluid dynamic, unsteady heat release and acoustic properties of the combustion chamber. In order to improve the understanding of stability properties in such complex systems encountered in many industrial applications, the flame structure of an atmospheric swirl stabilized burner of 30 - 75kW was investigated as a function of total mass flow rate, mixture temperature, air/fuel equivalence ratio and combustion chamber length. This parametric investigation revealed the existence of several flame types according to air/fuel equivalence ratios, mixture temperatures and combustion chamber length.

Transitions between flame states were sometimes smooth and continuous and, in some particular conditions, took place abruptly. The smooth transition coincided with isolines of the calculated Damköhler number while the sudden transition was associated with adiabatic flame temperatures. The flame types greatly influenced the pressure drops across the flame and the stability properties of the system. Unstable flames coincided with the large pressure drops measured across the flame. Moreover, acoustic measurements identified several unstable thermoacoustic modes with frequencies ranging from 200 Hz for the dominant mode, up to several kHz for the high frequency ones. The relative amplitude of the latest increased for very lean and rich flames. Finally, acoustic measurements in the non-reacting flow for similar conditions revealed the existence of two hydrodynamic modes, which characteristic frequencies scaled with the average velocity at the burner outlet.

Phase-averaged OH chemiluminescence pictures were obtained to visualize the characteristics of the flame along with thermoacoustic instabilities. The influence of the air/fuel equivalence ratio, the mixture temperature as well as the air mass flow rate on the flame shape, including the intensity of the reaction zone, the flame front, the position of the flame front and its movement within a period of oscillation were systematically investigated and assigned to the individual flame types. Additionally, a setup for the Planar Laser Induced Fluorescence (PLIF) diagnostic was built up enabling the correction of single-shot pictures for laser shot-to-shot intensity variations, laser sheet inhomogeneity and laser light absorption due to the investigated OH radical and the optical arrangement of the test facility. Furthermore, methane far-infrared absorption and an acetone-PLIF technique were used in the upstream and downstream section of the
combustor to determine the role of air/fuel equivalence ratio fluctuations and possible variations of the power density as a driving mechanism of thermoacoustic instabilities.

The oscillations of the heat release rates were stabilized by a secondary fuel injection forced with a giant magnetostrictive actuator. Beside other elementary techniques (e.g. measurement of the mechanical transfer function), the frequency response of this high speed valve was additionally investigated with acetone-PLIF by seeding the natural gas flowing through the valve with acetone. Phase-averaged pictures of acetone were then acquired to measure the fuel flow modulation induced into the combustion chamber by the secondary injection as a function of excitation phase. Additional fuel line dynamics and delays by mixing processes (including a convective time lag) of the gas with the air as well as attenuation of the modulated gas flow were identified. The basic performance of the actuation system used to modulate the secondary fuel injection, its influence on the reaction zone and the potential of NOx-formation were elaborated.

For the assessment of the functional efficiency of the secondary fuel injection strategy, all relevant flame types of the combustion system were attempted to stabilize with a simple phase-shift control algorithm. The parameter of the controller were tuned to achieve the best compromise between the overall reduction of the sound pressure level in the combustion chamber in conjunction with maintaining the formation of nitrogen oxide (NOx) at a low level. The control strategy using the secondary fuel injection is able to reduce the sound pressure level for all investigated operating conditions, including stable and unstable flames. In particular for the most unstable flames, the controller exhibits an excellent potential to reduce pressure oscillations (maximum reduction of the average (rms) sound pressure level of -18 dB) by mainly affecting the fundamental oscillations in the combustion system (maximum reduction of the peak sound pressure level of -45 dB, rise of the NOx-level <1 ppmv).
Zusammenfassung


Phasengemittelte Bilder der OH-Chemilumineszenz wurden aufgenommen, um die Charakteristik der Flamme zusammen mit thermoakustischen Instabilitäten zu visualisieren. Der Einfluss des Luft/Brennstoffverhältnisses, der Gemischt-temperatur, sowie des Massenstromes auf die Erscheinungsformen der Flammen zusammen mit ihren unterschiedlichen Intensitäten und ihrer Auswirkungen auf die jeweilige Flammenfront mit der zugehörigen Position und Bewegung wurde systematisch untersucht und einem spezifischen Flammentyp zugeordnet. Zusätzlich wurde ein Aufbau für die planare laserinduzierte Fluoreszenzdiagnostik (PLIF) erstellt, welche die Korrektur der single-shot Bilder bezüglich variierender Laserintensität, inhomogenem


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Nomenclature

This list of symbols is not exhaustive and the symbols that are not mentioned below are explained when they appear.

**Latin letters**

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Excited state of an electron</td>
<td>[-]</td>
</tr>
<tr>
<td>A</td>
<td>Parameter of a fitting technique (amplitude)</td>
<td>[V], [-]</td>
</tr>
<tr>
<td>$A_{forcing}$</td>
<td>Forcing amplitude of the GMM valve</td>
<td>[V]</td>
</tr>
<tr>
<td>$A_n$</td>
<td>Coefficients of the Fourier series expansion</td>
<td>[-]</td>
</tr>
<tr>
<td>arg { $G(j\omega)$ }</td>
<td>Phase response of a transfer function $G(j\omega)$</td>
<td>[rad], [°]</td>
</tr>
<tr>
<td>B</td>
<td>Parameter of a fitting technique (offset)</td>
<td>[V], [-]</td>
</tr>
<tr>
<td>c</td>
<td>Speed of light</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$C_{Absorption}$</td>
<td>Calibration factor for the absorption</td>
<td>[-]</td>
</tr>
<tr>
<td>$c_p(T)$</td>
<td>Specific heat</td>
<td>[J/(kgK)]</td>
</tr>
<tr>
<td>$C_i$</td>
<td>Calibration matrix to correct stripes of the images</td>
<td>[-]</td>
</tr>
<tr>
<td>$C_{int}$</td>
<td>Calibration factor of intensity variations of the laser</td>
<td>[-]</td>
</tr>
<tr>
<td>$c_{VtoPa,dB}$</td>
<td>Calibration factor for microphones</td>
<td>[Pa/V]</td>
</tr>
<tr>
<td>$c_x$</td>
<td>Concentration of species x</td>
<td>[mol/L]</td>
</tr>
<tr>
<td>CH$_4$</td>
<td>Methane</td>
<td>[Vol.-%]</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon monoxide</td>
<td>[ppmvdry]</td>
</tr>
<tr>
<td>CO$_2$</td>
<td>Carbon dioxide</td>
<td>[Vol.%dry]</td>
</tr>
<tr>
<td>$d_{EV}$</td>
<td>Diameter of the EV combustor (50[mm])</td>
<td>[mm]</td>
</tr>
<tr>
<td>$D_a$</td>
<td>Damköhler number</td>
<td>[-]</td>
</tr>
<tr>
<td>$E_t$</td>
<td>Energy of a molecule due to spin and orbiting</td>
<td>[J]</td>
</tr>
<tr>
<td>$E_r$</td>
<td>Rotational energy of a molecule</td>
<td>[J]</td>
</tr>
<tr>
<td>$E_v$</td>
<td>Vibrational energy of a molecule</td>
<td>[J]</td>
</tr>
<tr>
<td>$F(J)$</td>
<td>Rotational energy of a molecule</td>
<td>[cm$^{-1}$]</td>
</tr>
<tr>
<td>f</td>
<td>Focal length of a lens</td>
<td>[mm]</td>
</tr>
<tr>
<td>$f_{forcing}$</td>
<td>Forcing frequency of the GMM valve</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_i$</td>
<td>Frequency (longitudinal direction)</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_m$</td>
<td>Frequency (azimuthal direction)</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_n$</td>
<td>Frequency (radial direction)</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_{res}$</td>
<td>Resonance frequency</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_{SNIP(x)}$</td>
<td>Filtered sample at position x with the SNIP-filter</td>
<td>[-]</td>
</tr>
<tr>
<td>$</td>
<td>G(j\omega)</td>
<td>$</td>
</tr>
<tr>
<td>$G(v)$</td>
<td>Vibration energy of a molecule</td>
<td>[cm$^{-1}$]</td>
</tr>
<tr>
<td>$C_{controller}$</td>
<td>Controller gain</td>
<td>[-]</td>
</tr>
<tr>
<td>h</td>
<td>Planck’s constant</td>
<td>[Js]</td>
</tr>
<tr>
<td>i, j</td>
<td>Position of the pixel</td>
<td>[-]</td>
</tr>
</tbody>
</table>
$I$ Actual intensity value \([W/m^2]\)

$J$ Rotational quantum numbers \([-]\)

$k, m, n$ Mode numbers \([-]\)

$k_x$ Absorption coefficient of the species \(x\) \([m^{-1}]\)

$l$ Length of the laser beam path \([m]\)

$L$ Length of the duct (combustor) \([m]\)

$m_{air}$ Air mass flow rate \([g/s]\)

$m_{gas}$ Gas mass flow rate \([g/s]\)

$m_{valve}$ Gas mass flow rate through the GMM valve \([g/s]\)

$m_{valve, norm}$ Standard mass flow rate of the GMM valve at full stroke \([g/s]\)

$NO_x$ Nitrogen oxide \([ppmv]_{dry}\)

$O_2$ Oxygen \([Vol.-%]_{dry}\)

$O_{controller}$ Controller offset \([-]\)

$O_{forcing}$ Forcing offset of the GMM valve \([V]\)

$p$ Harmonic pressure oscillation (dynamic pressure) \([Pa]\)

$p'$ Pressure fluctuation (dynamic pressure) \([Pa]\)

$p_{ref}$ Reference pressure \([Pa]\)

$p_{rms}$ Root mean squared pressure signal (rms SPL) \([dB \text{ re } 20\mu Pa]\)

$P$ Probability \([-]\)

$PID$ Signal of the photovoltaic infrared detector \([V]\)

$PM$ Photomultiplier signal \([V]\)

$R$ Radius of the duct (combustor) \([m]\)

$S$ Spin quantum numbers \([-]\)

$SO_x$ Oxides of sulfur \([ppmv]_{dry}\)

$St$ Strouhal number \([-]\)

$t$ Time \([s]\)

$t_{phase}$ Time delay of the triggering signal \([s]\)

$T$ Period time \([s]\)

$T_{f, ad}$ Adiabatic flame temperature \([K]\)

$T_{mix}$ Mixture preheat temperature \([K]\)

$u$ Exit velocity at the combustor \([m/s]\)

$UHC$ Unburnt hydrocarbons \([ppmv]_{dry}\)

$v$ Vibrational quantum number \([-]\)

$V$ Command signal \([V]\)

$x$ Position along x axis \([mm]\)

$X$ Ground state of an electron \([-]\)

$y$ Position along y axis \([mm]\)

**Greek letters**

$\Delta I_{abs}$ Intensity difference due to absorption \([W/m^2]\)

$\Delta p_{flame}$ Pressure drop of the flame \([Pa]\)

$\Delta p_{flow}$ Pressure drop due to the EV combustor \([Pa]\)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Δp_{tot}</td>
<td>Total pressure drop of the EV combustor</td>
<td>[Pa]</td>
</tr>
<tr>
<td>Δp_{valve, norm}</td>
<td>Norm pressure across the GMM valve</td>
<td>[bar]</td>
</tr>
<tr>
<td>Δτ_{controller}</td>
<td>Time lag element of the controller</td>
<td>[-]</td>
</tr>
<tr>
<td>Δφ_{phase}</td>
<td>Difference of phase angles (intensity and position)</td>
<td>[°]</td>
</tr>
<tr>
<td>ΔΦ_{trig}</td>
<td>Triggering signal with respect to the phase angle</td>
<td>[V]</td>
</tr>
<tr>
<td>Δφ_{trig}</td>
<td>Phase angle of the triggering signal</td>
<td>[°]</td>
</tr>
<tr>
<td>λ, Σ, Φ</td>
<td>Quantum numbers</td>
<td>[-]</td>
</tr>
<tr>
<td>λ</td>
<td>Wave length</td>
<td>[m]</td>
</tr>
<tr>
<td>λ_{afr}</td>
<td>Air/fuel equivalence ratio</td>
<td>[-]</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
<td>[kg/m³]</td>
</tr>
<tr>
<td>τ_{flow}</td>
<td>Hydrodynamic time scale</td>
<td>[s]</td>
</tr>
<tr>
<td>τ_{chem}</td>
<td>Chemical time scale</td>
<td>[s]</td>
</tr>
<tr>
<td>ν</td>
<td>Frequency of the light source</td>
<td>[s⁻¹]</td>
</tr>
<tr>
<td>φ</td>
<td>Parameter of a fitting technique (phase)</td>
<td>[rad], [°]</td>
</tr>
<tr>
<td>φ_{trig}</td>
<td>Actual phase angle</td>
<td>[°]</td>
</tr>
<tr>
<td>ω</td>
<td>Angular frequency</td>
<td>[rad]</td>
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**Abbreviations**

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<th>Definition</th>
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<tr>
<td>ACC</td>
<td>Active combustion control</td>
</tr>
<tr>
<td>a.u.</td>
<td>Arbitrary unit, [a.u]</td>
</tr>
<tr>
<td>CCD</td>
<td>Charge coupled device (of a camera)</td>
</tr>
<tr>
<td>CLD</td>
<td>Chemiluminescence detector (gas analyzer)</td>
</tr>
<tr>
<td>CNG</td>
<td>Compressed natural gas</td>
</tr>
<tr>
<td>CoG</td>
<td>Center of gravity of an image</td>
</tr>
<tr>
<td>EV5</td>
<td>Dry low-NOx combustor with a diameter at exit of 50[mm]</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier transform</td>
</tr>
<tr>
<td>FID</td>
<td>Flame ionization detector (gas analyzer)</td>
</tr>
<tr>
<td>GMA</td>
<td>Giant-magnetostrictive actuator</td>
</tr>
<tr>
<td>GMM</td>
<td>Giant-magnetostrictive material</td>
</tr>
<tr>
<td>GT</td>
<td>Gas turbine</td>
</tr>
<tr>
<td>ICCD</td>
<td>Intensified charge coupled device (of a camera)</td>
</tr>
<tr>
<td>LAV</td>
<td>Aerothermochemistry and Combustion Systems Laboratory</td>
</tr>
<tr>
<td>LCC</td>
<td>Long combustion chamber (baseline condition)</td>
</tr>
<tr>
<td>LIF</td>
<td>Laser induced fluorescence</td>
</tr>
<tr>
<td>MIC1, MIC2</td>
<td>Microphone No.1, 2 at a fixed position downstream</td>
</tr>
<tr>
<td>MIC3, MIC4</td>
<td>Microphone No.3, 4 at a fixed position upstream</td>
</tr>
<tr>
<td>MPA</td>
<td>Magneto pneumatic gas analyzer</td>
</tr>
<tr>
<td>NDIR</td>
<td>Non-dispersive infrared gas analyzer</td>
</tr>
<tr>
<td>PID</td>
<td>Photovoltaic infrared detector</td>
</tr>
<tr>
<td>PLIF</td>
<td>Planar laser induced fluorescence</td>
</tr>
<tr>
<td>PM</td>
<td>Photomultiplier</td>
</tr>
<tr>
<td>PMT</td>
<td>Photomultiplier tube</td>
</tr>
</tbody>
</table>
Parts per million per volume
Reference pressure signal at a rms level of 20[µPa]
Root mean squared value
Region of interest (ROI0,ROI1,ROI2,ROI3)
Short combustion chamber
Statistic-sensitive non-linear iterative peak-clipping algorithm
Sound pressure level in [dB re20µPa], peak to peak or rms value
Transistor-transistor-logic, square signal used for triggering purposes

Subscripts

0 Reference value or value at a standard condition
cal Signal emitted by a calibrator
DI Dark field image
downstream Within the downstream section
dry Value expressed at a dry reference
diss Value of a specific emission gas
exit Value obtained at the exit
backcorr Resulting image corrected with a background image
BG Background image
corr Final image resulting after all transforms applied
Illuminated field image
norm Intensity value normalized with respect to a specific intensity value
S,in Signal stemmed from the beam splitter
S, out Signal after combustor
mean Mean value
norm Values normalized with respect to a specific intensity value
normalized Values normalized with respect to a specific reference
phaseing With respect to the actual phase
ref Reference value
ref 15% oxygen Value referenced to a 15% oxygen basis
rms Root mean squared value
fit Parameter of a sinusoidal fit
tot Total amount of a value
upstream Within the upstream section

Symbols

1a, 1b Flame type of the LCC and SCC (weak, lean)
2a Flame type of the LCC and SCC (after the fast transition)
2b Flame type of the LCC and SCC (slow transition, unstable)
3a Flame type of the LCC and SCC (rich, powerful)
3b Flame type of the SCC (rich, unstable)
IV Flame type of the LCC (rich, not investigated)
1 Introduction

1.1. Motivation

Continuous combustion processes are encountered in many applications related to power generation, propulsion systems, heating as well as domestic and industrial burners. The concept of lean premixed combustion has become widely accepted over the past twenty years, particularly in gas turbines, as an effective means to meet the stringent environmental standards of \( \text{NO}_x \) emission [56]. However, these processes exhibit a wide range of dynamics, thereby promoting the resonant coupling between unsteady combustion processes and pressure fields in the combustion chamber, leading to sustained large amplitude oscillations known as thermoacoustic instabilities [84]. These system instabilities enhance heat transfer at the combustor wall, deteriorate combustion efficiency and increase pollutant emission. In extreme cases, severe structural damage leading to the loss of control of the power plant or propulsion system can occur.

While perturbations of heat release are inevitably present in combustion processes and contribute to the broadband spectrum of the pressure fluctuations, they must be part of a self-exciting mechanism in order to initiate thermoacoustic instability. Perturbations in the system (fluctuation of velocity, fuel concentration, pressure, etc.) can cause heat release perturbations and consequently pressure waves, which must feed back energy into the initial perturbation to sustain the instability mechanism. While the global mechanism of thermoacoustic instabilities is known, identifying the initial mechanism responsible for it remains a difficult task. Indeed, the broadband spectrum of continuous combustion systems can trigger simultaneously several physically distinct mechanisms susceptible to initiate thermoacoustic instabilities. Consequently, such systems usually exhibit several typical time scales, each of them associated with different sources and feedback processes.

On one hand, lean premixed combustion offers the advantage of low \( \text{NO}_x \) emission but, on the other hand, it is highly sensitive to variations in equivalence ratio of the mixture that enters the combustion chamber. The response of the reaction rate to mix-
ture fluctuations, especially for systems close to the lean extinction limit, is significant. Such mixture modulations can be induced by unintentional interactions of pressure and flow oscillations with the reactant supply rate or even contrariwise be intended by a secondary fuel injection. Control strategies using a secondary fuel injection system could be able to suppress thermoacoustic instabilities over a wide range of frequencies by breaking the feedback process apart resulting in a stable heat release rate. Moreover, active combustion control (ACC) affords a better manageability of the combustion process and has been achieved in many combustion instability problems. Experiments also indicated that active control had potential applications in combustion enhancement ([13],[18]).

In contrary, passive control (i.e. design changes) does in general not provide a means for controlling these instabilities due to the multiple numbers of modes that may be excited by hydrodynamic or combustion processes.

1.2. Outline of the thesis

A straightforward identification of the origin of thermoacoustic instabilities remains difficult. Modifications of the combustion chamber geometry and comparing measurements for similar flows with (and without) combustion might provide insight into the origin of unstable modes. To this end, a detailed understanding of the dominant fluid dynamic and thermochemical driving mechanisms is absolutely necessary in order to prevent and to control combustion related instabilities.

For this purpose, an atmospheric lab burner facility (Chapter 3) was built up and specifically adapted measurement instrumentation were implemented.

In Chapter 4, the flame types observed have been characterized by different flame shapes, flame locations and sound emission. Transitions between different flame types were sometimes smooth and continuous ("slow transition") and, in some particular conditions, took place abruptly following a very small change of one of the parameters ("fast transition"). A similar behaviour was previously observed experimentally and is reported in ([11],[44]).

The first task of the current investigation consisted thus in identifying flame structures and transitions, in the space defined by the air/fuel equivalence ratio, mixture temperature and total mass flow rate. This investigation was performed for two combustion chambers of different lengths, and flame transitions were compared with constant isolines of adiabatic flame temperatures and Damköhler numbers. Pressure drops across the flame, pressure fluctuations in the combustion chamber and OH-Chemiluminescence, recorded with a photomultiplier, were measured for all conditions and both geometries. In addition, the frequency content of the dynamic pressure has been calcu-
lated for all flame types and the dominant modes were identified and analyzed as a function of air/fuel equivalence ratios and mixture temperatures. Finally, acoustic properties of reacting and non-reacting flows were compared (Chapter 4).

In addition to the acoustic characterization of the combustion chamber, the flame structure was derived with different flame visualization techniques (Chapter 5). The methods employed include the visualization of OH radicals as a tracer of the reaction zone (Chemiluminescence and laser induced fluorescence), the visualization of air/fuel equivalence ratio fluctuations through a dopant based laser induced fluorescence and a $CH_4$ absorption technique. The mean flame position, the motion of the flame front and the air/fuel equivalence ratios variations regarding to the pressure fluctuations have been evaluated and major differences between stable and unstable operating conditions could be observed.

The actuator used within the flame stabilization concept (Chapter 6) was tested in detail, e.g. the transfer function between input signal and output response, the fuel injection properties, the fuel mixing characteristics and the potential of $NO_x$ formation due to a secondary fuel injection were classified. Within a next step, the flame positions, the amplitude of the pressure oscillations and the combustion pollutants of the controller stabilized flames were reinvestigated and compared to the unstable cases.

The final section (Chapter 7) will summarise the conclusions drawn from the thesis, the main achievements of the work, and prospects for future secondary fuel injection based on control strategies applied to lean combustion processes in gas turbines.

Since the present work is conducted within the framework of larger project on thermoacoustic instabilities investigated at the ETH Zürich (in collaboration with Alstom Power, Switzerland), it provides a basis for several other advanced research activities. The control oriented and more theoretical work of André Niederberger affiliates to this work as logical consequence ([68],[69]). Within his complementary work, the combustor is split into different elements deriving an acoustic network. Its parameters were found analytically and the elements covering the flame, the combustor and a source term were experimentally identified for several operating conditions. Beside passive control aspects, the work investigates basic considerations of the control of thermoacoustic instabilities in combustion systems. Moreover, several controller algorithms were developed (Phase-shift controller, $H_\infty$-controller and a genetic algorithm, provided by N. Hansen, ICOS, ETH Zürich). In addition to the control strategy using a secondary fuel injection, the actuation with loudspeaker is supplementary discussed in more detail.
Outline of the thesis

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This chapter provides an overview of the early history of thermoacoustic instabilities (Higgins, Rayleigh) and introduces its simplest experiment (Rijke tube), makes a general survey of their occurrences in industrial applications, states the reasons for the sensitivity of the actual gas turbine technology to combustion oscillations, gives a broader introduction to thermoacoustic instabilities in lean premixed combustion and finally summarises actuation concepts to stabilise the combustion instabilities, especially emphasizing the concept of the secondary fuel injection.

The research activity in the field of thermoacoustic instabilities in the past century was immense and literature concerning thermoacoustic instabilities is abundantly available. However, the literature referred to within this chapter is not exhaustive in the sense of a complete documentation of the phenomenon “Thermoacoustic Instability” - it rather tries to summarise all relevant and preparatory research in the context of this work.

2.1. Rayleigh’s criterion

The age of thermoacoustic instabilities starts in the year of 1777 with the first observation of “singing flames” by Higgins (reported 1802 in [41]). He was able to excite a characteristic sound wave by placing a hydrogen diffusion flame into a tube. More than a century later (1878), Lord Rayleigh could scientifically characterize the occurrence of thermoacoustic instabilities by specifying a necessary condition of such instabilities by describing a mechanism where heat release excites pressure oscillations [83]. He described thermoacoustic instabilities as “the most interesting examples of vibrations maintained by heat” (Rayleigh, 1948, p. 226, [84]) in the following way:

“If heat be periodically communicated to, and abstracted from, a mass of air vibrating (for example) in a cylinder bounded by a piston, the effect produced will depend upon the phase of the vibration at which the transfer takes place. If heat be given to the air at the moment of greatest condensation, or taken from it at the moment of greatest rarefaction, the vibration is encouraged. On
the other hand, if heat be given at the moment of greatest expansion, or abstracted at the moment of greatest condensation, the vibration is discouraged." (Rayleigh, 1948, p. 226, [84])

Even today, Lord Rayleigh’s famous criterion is used by many researchers to characterize these heat driven oscillations. Lord Rayleigh himself stated the “principal question” to be considered to encourage the feedback process as the phase relation between the heat release relatively to the oscillation of the acoustic pressure.

In 1954, the first mathematical formulation of the Rayleigh criteria was derived by Putnam and Dennis [81], whereby its integral form is commonly used.

\[
R = \int_0^T p'(t)q'(t)\,dt > \text{dissipation losses} \tag{2.1}
\]

The symbol \( \cdot \) denotes fluctuating values of the pressure \( p \) and heat release rate \( q \) whereby these values integrated over a period of the instability \( T \) result in the Rayleigh index \( R \). A positive value of \( R \) stands for an energy transfer from the heat release to the pressure oscillations (amplification of the pressure oscillations due to the fluctuating heat release) and a negative Rayleigh index corresponds to a damping of the pressure oscillation (indicating an inverse energy flow). In any case, the coupling between the heat source and the pressure wave is essential for the formation of self-sustained thermo-acoustic instabilities.

The most recent work by Nicoud and Poinsot [67] proposes an extension of the standard Rayleigh criterion by including entropy changes in addition to the velocity and pressure fluctuations.

2.2. The Rijke tube

Probably the best explored thermoacoustic phenomenon, experimentally as well as theoretically, is the experiment investigated originally by Rijke [89] in the year of 1859. The so-called Rijke cylindrical duct is shown in Figure 2.1 with the optimal location of the heat source at \( L/4 \), typically a heated gauze. Since both ends of the vertical tube are open, a convective flow in the upwards direction is established due to the temperature source. The pressure nodes of the 1st axial mode are located at the open ends of the tube. At the same positions exist antinodes of the velocity fluctuation. Though containing less energy and thus not shown here, higher modes are present in the tube as a matter of course. The buoyancy driven flow, although in principle a steady flow, is important for the feedback process. Any presence of a pressure oscillation, e.g. an arbi-
Figure 2.1. A schematic of a Rijke tube (left) depicted with the shapes of the first pressure and velocity modes and the corresponding Rayleigh index (right) with a neglected time delay between the velocity fluctuation and the heat release rate (the heat source is moved along the vertical axis to evaluate the Rayleigh criterion).

Rarity initial perturbation, causes its related velocity fluctuation and this oscillation itself is superimposed on the steady velocity field due to the buoyancy effect of the heat source. As a consequence, the unsteady flow across the heated gauze results in a fluctuating heat release (heat release follows velocity fluctuations with a specific time delay, \( q'(t) \sim u'(t - \Delta t) \)) which in turn creates a pressure fluctuation and the geometry of the duct responses at its characteristic frequencies (depending on the properties of the fluid, the mean inlet velocity, etc.). The established pressure oscillations set the velocity oscillations and so forth.

If the heat source is moved away from its optimal place, the coupling between the pressure oscillations and the heat release rate is getting less effective to excite itself and the relative amount of energy consumed within the feedback process increases till the self-excited oscillations completely disappear or even is damped. By evaluating the Rayleigh integral, the statement of Lord Rayleigh defining the phase relation as principle relationship became obvious.

For the lower part of the tube, the phase of the velocity is 90° ahead of the phase of the pressure and taking a phase delay of a few degrees of the heat release (with reference to the phase of the velocity since the heat transfer is directly coupled with the flow velocity) into account, the Rayleigh index is still positive and oscillations were self excited. If the upper part of the tube is heated, the phase of the velocity lays 90° behind pressure
Industrial applications and thermoacoustic instabilities

Oscillations and corresponding heat release oscillation is, regarding to the pressure oscillation, even more delayed. This results in a negative Rayleigh index and the entire system is damped by the heat source.

Another well-known as simple example of an experiment owning self-excited oscillations is the Sondhauss tube [102]. It is basically a “modified” Rijke tube with a hot bowl at one of its end and was discovered in the glass making by glass-blowers.

The inversion of a Rijke tube was reported by Bosscha und Riess ([87], [88]). They placed a cold grid in the upper part of a Rijke tube and thereby gave rise to pressure oscillations. The heat sink, in contrary placed in the lower part, involves a damping of the system by transferring acoustic energy out of the acoustic system.

2.3. Industrial applications and thermoacoustic instabilities

In many applications, thermoacoustic plays an important role within occurring physical processes. However, not all of thermoacoustic oscillations are unintentional since some of them were deliberately deployed for a specific industrial application. The following listing gives a short overview of the history of thermoacoustic evolved over the last decades.

- 18th / 19th century: Lord Rayleigh’s discoveries and, in different ways, the Rijke tube was dominating the scientific work related to thermoacoustic instabilities. In particular, researchers put effort in the causes of such acoustic oscillations.

- In the 1950’s, thermoacoustic instabilities appear in solid and liquid propellants rocket engines ([16], [79]). As a cause of the armament during the cold war, research activities in this domain are surged to overcome the problems of thermoacoustic instabilities to prevent large amplitudes of pressure oscillations. Even in the 21st century, continuative topics are still being researched.

- In the 1960’s, acoustic heat engines were developed for the first time. The basic principle for such machines is that heat, flowing from a temperature source to its sink, produces acoustic energy whereby this acoustic energy itself can be transduced to electric power (See Swift for a review of thermoacoustic engines [103]). In the same period, the first machines, intentionally using thermoacoustic oscillations in an “inverse matter”, were the pulsed tube coolers invented by Gifford and Longsworth [32]. Acoustic energy introduced into a system is used to force a heat flow from its sink to the source and by doing so, the temperature at the sink is lowered [76].

- After the 1970’s, the industry using boilers and furnaces faced thermoacoustic problems as well as plants widely used in the domestic area for heating, ventilation and air conditioning (e.g. see Putnam’s book [80]).
• In the 1980’s, the ramjet technology, an adapted propulsion system for high supersonic flight, gave the research field of thermoacoustic instabilities an additional boost. Moreover, the request for higher flight velocities could be fulfilled with afterburners even for turbine engines (Aeroengine afterburners). Both techniques are susceptible for thermoacoustic instabilities and consolidated knowledge of the occurrences of low frequency pressure oscillations has been acquired over the past years (e.g. see the “reheat buzz” of Bloxsidge et al. [10]).

Another technology, called pulsed combustion, mainly developed within this period (e.g. by Keller et al. ([6], [46]) at the Sandia National Laboratories, Livermore, USA), profits of the existence of thermoacoustic instabilities. The instability forces, a better mixing of the fuel with the combustion air, increases the burning rate of heavy liquid fuels, and therefore improves the combustion of heavy fuels and results in a mellower emission characteristic [109].

• Already starting in the 1990’s, stationary gas turbines manufacturers were exposed to stringent emission regulations and felt impelled to cut down NOₓ emissions by re-designing their combustion chambers. In particular, the dry-low-NOₓ-combustor technology encountered major difficulties with thermoacoustic instabilities. Furthermore, the aviation industry, beside the ground-based gas turbine manufacturer, is legally bound to reduce the NOₓ emission within the following years (superiorly without making jet engines more prone to thermoacoustic instabilities).

Figure 2.2 gives an overview of the development of the NOₓ emission produced with the state of the art technique. Environmental concerns started to have an impact on the technology in the late 1970’s. The maximum temperature of the combustion process (usually a diffusion flame) was reduced with the use of water (or steam) injected into

Figure 2.2. Development of the NOₓ emissions of Gas turbines over the last decades and improvements of the efficiency of power generation in GT/CC power plants [7].

9
the gas turbine (before the combustor) and as a consequence, the formation of $NO_x$ was significantly lowered. This concept was able to fulfill the first requirements of the EPA (U. S. Environmental Protection Agency) legislation standards (set to a $NO_x$-level of $75 \text{ ppmv}_{dry, \text{ ref } 15 \% \text{ oxygen}}$). In 1983, the BACT legislation became effective (California, United States), meaning that proposed facilities set as major polluting facility have to use the Best Available Control Technology (BACT) to fulfill this legislation without causing any severe economic drawback by running the facility (plant).

However, a competitive run of the gas turbine manufacturer was started to lower the $NO_x$-levels and to reduce the $CO_2$ output by increasing the efficiency of the power plants. As a result, dry-low-$NO_x$-combustors using premixed combustion were developed. By using this technology, emission values fell easily below the old EPA standards without any water injection to lower the peak temperature of the combustion or any need of an after-treatment of exhaust gases. Such burners, all having very low raw $NO_x$ emissions in the range of $10 - 25 \text{ ppmv}_{dry, \text{ ref } 15 \% \text{ oxygen}}$, are to a greater or lesser extent prone to thermoacoustic instabilities.

Other emissions, legislated of the BACT are the lead content, particulate matter (PM), sulfur dioxide ($SO_2$), carbon monoxide ($CO$) and the ground level ozone ($O_3$). The actual power plant (Nov., 2005), defining the best available control technique (BACT) for major polluting facilities with a gas turbine application ($>50 \text{ MW}$) is the power plant in Vernon, California [1]. The new combined-cycle power plant consists of $43 \text{ MW}$ gas turbine (Alstom GTX100, dry-low-$NO_x$-combustor EV, natural gas) and a $55 \text{ MW}$ steam turbine (start-up date: Fall 2004).

For further reduction of the raw $NO_x$-levels, a low temperature SCR catalyst (selective catalytic reduction) with aqueous ammonia (19% by weight) is used along with an oxidation catalyst to achieve lower $CO$-levels.

<table>
<thead>
<tr>
<th>Emission permit limit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$NO_x$, 1-hour average</td>
<td>$2 \text{ ppmv}_{dry, \text{ ref } 15 % \text{ oxygen}}$</td>
</tr>
<tr>
<td>$CO$, 3-hour average</td>
<td>$3 \text{ ppmv}_{dry, \text{ ref } 15 % \text{ oxygen}}$</td>
</tr>
<tr>
<td>$UHC$, 1-hour average</td>
<td>$2 \text{ ppmv}_{dry, \text{ ref } 15 % \text{ oxygen}}$</td>
</tr>
<tr>
<td>$NH_3$, 1-hour average</td>
<td>$5 \text{ ppmv}_{dry, \text{ ref } 15 % \text{ oxygen}}$</td>
</tr>
<tr>
<td>Particle matter</td>
<td>$0.01 \text{ [gr/ft]}$</td>
</tr>
</tbody>
</table>

Table 2.1. Permit limit of the Vermont power plant (current BACT determination)
2.4. Instabilities in lean premixed combustion

![Diagram]

Figure 2.3. Basic phenomena involved in the feedback mechanism affecting the onset of thermoacoustic instabilities in combustion processes.

The lean dry low NO\textsubscript{x} combustor type is widely encountered for land based gas turbines because of its excellent emission performance even though the advancement of lean premixed combustion is retarded by the presence of thermoacoustic instabilities. The main phenomena involved in the feedback process affecting the onset of thermoacoustic instabilities in combustion systems depicted in Figure 2.3 in a simplified way. However, all of these phenomena exhibit a wide range of dynamics and are able to promote a resonant coupling between unsteady combustion processes, pressure field in the combustion chamber, the mixing process of reactants and flow field. While perturbations of heat release are inevitably present in combustion processes and contribute to the broadband spectrum of acoustic frequencies, they must be part of a self-exciting mechanism in order to initiate thermoacoustic instability. Fluctuations of velocity and pressure field (acoustic oscillations), perturbations in the fuel concentration and in the mixing process of the burnt/unburnt gases can cause heat release perturbations and consequently pressure waves. With the correct phasing of these dynamic systems (corresponding time scales presumed), energy is fed back into the initial perturbation and sustains the instability mechanism (Rayleigh criteria). Notwithstanding the knowledge of the global mechanism concerning thermoacoustic instabilities, the identification of the initial mechanism which starts the process is still challenging. The broadband combustion noise and the resonant response at characteristic frequencies of the combustor geometry can trigger simultaneously several physically distinct mechanisms to set the snowball rolling.
Furthermore, a complete understanding of thermoacoustic instabilities does not only require the knowledge of the initiating mechanism, but also of the saturation mechanism. As shown by Fichera et al. [25] and Lieuwen et al. [58], characteristics of limit cycles observed in the laboratory for an unstable combustion regime are strongly influenced by non-linear processes saturating the mode’s amplitude.

Premixed combustion systems found in industrial or aeronautics applications involve a large number of physical processes playing a more or less determinant role for the overall stability of combustion processes. In high pressure systems such as rockets, pressure fluctuations from unsteady combustion can directly trigger heat release oscillations nearly in phase with the pressure wave [110]. In liquid fuel combustion, fluctuations of pressure or velocity can affect combustion processes by altering the path of the flow of the injected fluid or by changing the spray break-up process. The resulting change in combustion rate might be sufficiently large and might occur at the proper time to drive pressure oscillations [80]. For combustion systems burning gaseous reactants (usually associated with flowing media), pulsations in supply rate, variations of flame front area, vortex shedding activity ([31],[74],[77],[94]), fluctuations of mixing rate between burnt and unburnt mixtures and periodic composition change ([53],[54],[59]) have been identified as mechanisms able to initiate thermoacoustic instabilities. While supply rate oscillations can be partially suppressed or minimized by providing a large pressure drop through the supply system, vortex shedding activity is mainly controlled by velocity and density profiles at the burner exit [65] as well as flame position [28]. These characteristics are in most cases directly involved in the flame stabilization mechanism and thus the room for manoeuvre is limited in technical applications.

Hydrodynamic instabilities ([31],[74],[77],[94]) and air-to-fuel modulations ([70],[95]) have been given lots of attention in the literature as potential candidates for driving instabilities in various gaseous burner configurations. The driving source and feedback processes both differ for each mechanism and might exist for confined and non-confined flames. However, acoustic boundaries of confined flame facilities passively but selectively modulate the amount of energy fed back into the initial perturbation and thus modify its growth. An efficient feedback loop only takes place around discrete frequencies given by acoustic eigenmodes of the combustion chamber. Consequently, the frequency of thermoacoustic modes observed for confined systems usually “clumps” near the natural acoustic frequencies of the system, constituting one of the few characteristics common to all thermoacoustic instabilities observed in confined facilities.

Hydrodynamic instabilities in gaseous burners have been extensively investigated experimentally ([31],[74],[77],[94]) and numerically ([2],[24],[47],[105]). For exam-
Instabilities in lean premixed combustion

The experimental work of Poinset et al. [77] showed that low frequency instabilities found in a multiple-flame-holder dump combustor occurred at the acoustic eigenfrequencies of the system. Although this frequency did not correspond to the most amplified shear layer instability mode found for non-reacting jets [43], the acoustically-induced velocity fluctuations at the burner exit were sufficient to trigger vortex shedding and thus to periodically increase/decrease fine scale mixing between burnt and unburnt mixture in the vortices. The large periodic heat release associated with the burning inside the vortices provided the necessary feedback processes to make the initial pressure perturbation grow further. Similar observations were made by Schadow and Gutmark in a single dump combustor [94]. In this investigation, vortex merging during convection from the dump plane to the flame location was considered as the mechanism responsible for the lower frequency of thermoacoustic instability compared to the preferred mode of shear layers. Paschereit et al. [74] measured eigenmodes of the axial velocity perturbation, in a water channel, at the dump plane of a swirl burner similar to the one used in the current investigation. They observed three dominant hydrodynamic modes: an axisymmetric and a helical mode both associated with the jet like profile (external shear layer) at non dimensional frequencies of 0.58 and 7.7 respectively and one helical mode associated with the wake like profile at a non dimensional frequency of 1.16. These measured frequencies were supported by a linear stability analysis performed on experimental velocity profiles.

The influence of swirl on oscillations in ducted flames was experimentally investigated by Sivasegaram and Whitelaw [101], who systematically studied flames stabilized behind a disk, a reward-facing step and an annular ring. In the case where swirl increased the axial distance over which heat release took place (flame stabilized behind a disk), oscillations were reduced due to a weaker correlation between fluctuating pressure and heat release. On the contrary, swirl shortened the flame length for flames stabilized behind a sudden expansion and thus increased oscillations, while flames behind the annular ring stayed unchanged. Aerodynamic characteristics not only play a determinant role in flame stabilization and hydrodynamic stability, but also in spatial dispersion of equivalence ratio or entropy waves as they are convected downstream from the fuel-oxidizer mixer down to the flame and combustor exit. In this context, Sattelmayer [92] showed that spatial dispersion of the flow plays a crucial role in equivalence ratio waves.

The well-known Kelvin-Helmholtz like instability, associated with a periodic redistribution of vorticity in shear layers, is not the only flow-driven process able to induce periodic variations of the local mixture composition. The flow pattern in swirling flows often exhibits the three-dimensional time dependent instability called the Precessing Vortex Core (PVC) ([22],[26],[27]). The PVC phenomena observed in flows with suf-
ficient swirl to induce vortex breakdown [22], generating back flows responsible for flame stabilization. The PVC is the consequence of a slight deviation between the axis of the central recirculation zone and the geometrical axis, inducing the precessing motion of the recirculation zone around the axis of symmetry. The normalized frequency of the precessing vortex core is a function of swirl number and is of the order of 0.6 for swirl numbers similar to those in the current investigation.

On one hand, lean premixed combustion offers the advantage of low $NO_x$ emissions but, on the other hand, is highly sensitive to variations in equivalence ratio of the mixture that enters the combustion chamber. The response of the reaction rate to mixture fluctuations, especially for systems close to the lean extinction limit, is significant and may lead to periodic extinction of combustion processes in extreme cases. Such mixture modulations can be induced by interactions of pressure and flow oscillations with the reactant supply rate. The fuel concentration perturbations are formed near the fuel injector and are convected by the mean flow to the flame where they produce large heat release oscillations. The dominant characteristic time associated with this mechanism is the convective time from the point of formation of the mixture to the point where it is consumed at the flame. It can be shown that instabilities take place when the ratio of this convective time to the period of the instability equals a constant specific to the combustor design ([57], [59]).

Apart from the well-known equivalence ratio fluctuations and vortex shedding activities, turbulent flame speed fluctuations may also cause heat release oscillations. Acoustic fluctuations may cause periodic changes of turbulent intensity which strongly affect the turbulent flame speed and thus induce reaction rate oscillations able to initiate thermoacoustic instabilities. This assumption was tested in an acoustic network model based on transfer matrices and compared with experimental data gathered for a premixed mode of combustion (no equivalence ratio oscillations) and showed very good agreement [95].

Shear driven instabilities and mixture modulations can coexist simultaneously in continuous combustion systems, leading to distinct modes of instabilities distributed over a wide range of frequencies (from hundreds to several thousands Hertz) monopolizing most of the acoustic energy. Physical and chemical key parameters of such aerothermochemical systems are strongly coupled and thermoacoustic modes are saturated immediately after flame ignition or after a slight change of one of the parameters.
2.5. Control of combustion instabilities

In recent years, there has been considerable activity addressing the control of thermoacoustic instabilities. The research effort, throughout investigations carried out on lab scale experiments and the succeeding theoretical analysis, has shown the potential to damp or even to control combustion oscillations. In principle, two main categories of the control of thermoacoustic instabilities exist in gas turbine combustors. The **passive control** strategy is widely used in almost every industrial combustor compared to the **active control** techniques.

Since acoustic wave propagation is part of the feedback mechanism involved in thermoacoustic instabilities, passive control designs resort often to modify the resonant system of the combustor [80]. Baffles, mufflers, Helmholtz resonators [8], acoustic liners and perforated plates were mounted at the combustor to change the boundary conditions and to damp acoustic attributes of the system. This design changes of the combustor result in a continuous combustion less prone to thermoacoustic instabilities. However, in general passive control does not provide a means for controlling these instabilities due to the multiple number of modes that may be excited within the combustor at different operating conditions. Since passive control methods are optimised for specific frequencies and thus, for specific operating conditions as well, the applied stabilizing concept shows a good performance for these design points only. In contrary, the passive control scheme is not able to wipe the oscillations out of the combustor for conditions not considered during the development time. This costly dry and error within the layout design and the missing adaption mechanism for any unforeseen event are major drawbacks of passive control strategies.

In contrary, control strategies using an actively actuation system to stabilize the combustion were able to suppress thermoacoustic instabilities over a wide range of frequencies and afford a better manageability of the combustion process. The control concept has to provide a robust operation of the combustor within several operating conditions (load, power density, transient operation, etc.), changed boundary conditions (work environment, fuel composition, etc.) and any problems stemmed from ageing effects of the combustor and the surrounding components. For these purposes, feedback control is the basic recommended way to control dynamic systems and its long-term changes in a sufficient way. This control strategy has been implemented in countless applications in almost every technical domain. For the field of combustion instabilities, the term Active Combustion Control (ACC) is widely used. Extensive literature on feedback control of combustion instabilities exists and is already exquisitely classified in the reviews by Docquier and Candel [18], Candel [14], McManus et al. [62] and Schadow and Gutmark [94] as well as by Dowling and Morgans [19].
Pressure transducers, located at a sensitive position of the combustion chamber as well as OH and/or CH chemiluminescence detectors are commonly used as monitoring devices of the combustion process and basically serve as possible input signals for the feedback controller. Although it is important for the controller to know the actual state of the combustion system (any dynamical information sufficiently characterizing the actual state), the detection component itself seems not to be the bottleneck of the control system. Pressure transducers and chemiluminescence detectors are accurate, provide a high frequency response and are applicable even in the rough environments of gas turbine combustors. More deciding far the overall performance of the controller is its actuation part. Even having the best detection device and controller algorithm available, the control system will fail without an actuator providing sufficient control authority along with minimised time delays to force the combustion process.

Although the development of controller-oriented models and the controller design itself are beyond the scope of the current work, the investigation of Annaswamy and Ghoniem [3], Campos-Delgado et. al. [13], Schuermans [96], and Riley et. al. [90] must be mentioned as examples of today’s control approaches related to lean premixed combustion.

Any strategy of an actuator has to break apart the feedback loop of mechanisms affecting combustion instabilities (See Figure 2.3). In principle, four types of actuators, more or less effective, exist to prise the loop open:

The first type of actuator used for active combustion control is adapted from the passive control strategies and similarly zero in on the modification of acoustic attributes of the combustor. The geometry of Helmholtz resonators, quarterwave tubes and/or boundary conditions of the in/outlet flow can be dynamically changed to tune the frequency range where they are absorbing acoustic oscillations. Although the dynamic frequency response of such actuators is permitted to be low (a few Hz will fully suffice to adjust the geometry of resonators), the implementation into real applications is difficult due to a low robustness (in a mechanical sense) as well as the enlarged space requirements of this resonators, especially to damp acoustic oscillations at lower frequencies.

A second type of actuator is intervening in the identical phenomena involved in the feedback mechanism of thermoacoustic instabilities as the previous type of actuator. Loudspeakers, primarily not designed for the changing of the boundary conditions, interfere directly with the acoustic field establishing in the combustion chamber. The active control of combustion instabilities by speaker actuation has been reported by Lang et al. [51], Campos-Delgado et al. [12], Schuermans ([96], [97]), Paschereit et al. [73] and by Niederberger ([68], [69]) for the adapted use within for the current test
Control of combustion instabilities

Typical frequency responses of loudspeakers (broadband) cover the entire audible range from 20[Hz] to 20[kHz]. However, a typical unstable combustion system exhibits the first longitudinal mode in the frequency range of 100 – 500[Hz] (depending on the geometric scale of the combustion chamber). So, loudspeakers adapted for combustion control are large woofers. The necessary space and size, the lacking robustness in the combustion environment (not in a control sense) and the small control authority due to the lack of inductable acoustic power into the combustion system, still avoid an application under real gas turbine conditions.

A third actuator strategy focuses on the possibility to modulate the flow characteristics before or while it enters the combustor or at least directly within the combustion chamber itself. The manipulation of the flow dynamics, e.g. the forcing of the initial shear layer by velocity perturbations results in small scale vortices and thus in stabilizing the entire combustion process. The control of individual (acoustic) modes within the combustion process is difficult since the receptivity of the shear layer is strongly related to the characteristics of the flow (jet). (See Gutmark for a review of the shear flow modulation in combustion control and McManus et al. [63] for the influence of the excitation of the shear layer on the combustion performance.)

Another method of modulating the flow has recently been presented by Uhm and Acharya [107]. They have controlled the first longitudinal mode of the combustion instability with an additional highly modulated high-momentum air jet by influencing the mixing of reactants within the reaction zone of the combustor. Additional investigations showed that by forcing the combustion with a low-frequency modulation (a few Hz) superimposed on the high-momentum air jet, the instability can be reduced in the order of one magnitude [108].

The fourth type of actuators used along with the control of thermoacoustic instabilities, is in principle based on an intended perturbation of the fuel rate by the main gas flow or by a secondary fuel injection to stabilize the heat release rate. Since the heat release rate and pressure waves are coupled, the frequency response of any fuel actuation system has to lie within the frequency range of pressure oscillations aimed to stabilize. In contrary to the loudspeakers, fuel actuation is limited in terms of the achievable frequency response of the gas flow modulation (the typical mechanical frequency response is <450[Hz]). Even if the modulated flow rate is very small compared to the main gas flow, the chemical energy released by this perturbed combustion process and the resulting marginal quota generated by an unsteady, fluctuating heat release and converted to acoustic energy, is in principle completely sufficient to cope with any acoustic oscillation of a gas turbine combustor.
Indeed, the secondary fuel injection sounds promising - but, there is a catch in it. The fuel actuator and the role of the position of the fuel injection are deciding for the overall performance of the control strategy applied for premixed combustion systems. If the secondary fuel is directly injected in the burning zone thus minimizing a convective time lag and maintaining a fast actuation response by the heat release, the risk of an insufficient mixture of the oxidant and the reactant readily causes a significant rise of the nitrogen oxide emissions and in the worst case, due to a locally changed reaction zone structure to a partially “diffusion-type”, produces even soot in the exhaust emission. In contrary, injecting the secondary gas flow in a zone upstream of the burner, guarantees a good mixing of the fuel with the oxidant, but increases the transport lag of the gas modulation and attenuates the modulation, both resulting in a weaker frequency response of the actuation system.

Even though many investigations used the secondary fuel injection in their control concepts, little research has been made with respect to an accurate location of the injection itself and the associated effects (actuator performance versus pollutant emission). In addition, most actuators use liquid fuels to operate, thus providing an improved frequency response compared to gaseous working fuels, the additional effect of atomizing and evaporating the liquid jet prior to the main reaction zone complicates the development of an accurate injection position.

Beside the acoustic excitation of the fuel line by loudspeakers, the intended modulations of the main or of an additional, secondary fuel flow were primary implemented by using distinct types of valves. The first valves used for combustion control in tube combustor were automotive fuel valves exhibiting an unwanted on/off characteristic (Langhorne et al. [52]). The dissipation within the flow and the mixing process smoothed most of the square-shaped injection profile, thus acceptable results were obtained. The design concepts of the subsequent investigations were realised with several types of solenoid valves, also exhibiting an on/off characteristic. These outcomes were generally improved with proportional solenoid valves whereby direct driven valves (produced by MOOG Inc.) were leading the competition compared with other servo valves. Servo valves are always driven by a (linear) motor whereby the masses of the moving parts of the motor and the spool are the limiting factor restricting the maximal frequency response. Additionally used to modulate a flow are piezoelectric stacks placed in the fuel supply line and varying the outlet area as well as magnetorestrictive driven valves. Both materials, piezoelectric and magnetorestrictive, though owning an extraordinary high frequency response, were slow in demand for active combustion control due to their high voltage requirements and the small and non-linear magnitude response of the material.
Table 2.2 summarizes the investigations and recent work done in the field of active combustion control and the achieved results of the control strategy. The listing is restricted to control applications solely using gaseous fuels and itself chronologically listed according to the date of publication.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Working Gas</th>
<th>Type of Actuator</th>
<th>Actuator Response</th>
<th>Peak-Level (p') Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Neumeier and Zinn [66], 1996</td>
<td>Methane</td>
<td>Magnetostrictive</td>
<td>400 [Hz]</td>
<td>-26 [dB] at 370 [Hz]</td>
</tr>
<tr>
<td>Rocket combustor</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No emissions measured</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lacy et al., 1998 [50]</td>
<td>CNG</td>
<td>Servo</td>
<td>200 [Hz]</td>
<td>-18 [dB] at 260 [Hz]</td>
</tr>
<tr>
<td>Additional air flow modulation</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seume et al. [100] (1998)</td>
<td>CNG</td>
<td>Servo</td>
<td>350 [Hz]</td>
<td>-20 [dB] at 145 [Hz]</td>
</tr>
<tr>
<td>Hermann et al. [39] (1999)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Modulated pilot flame (diffusion)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Full scale application: Siemens Vx4.3A and other GTs</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Richards et al. [85], 1999</td>
<td>CNG</td>
<td>Solenoid</td>
<td>20 [Hz]</td>
<td>-11 [dB] at 300 [Hz]</td>
</tr>
<tr>
<td>Low-frequency modulation</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Paschereit et al. [72], 1999</td>
<td>CNG</td>
<td>Servo (MOOG)</td>
<td>350 [Hz]</td>
<td>-20 [dB] at 0.54 &lt; Str &lt; 0.70</td>
</tr>
<tr>
<td>Emission + OH measurements</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Atmospheric test rig of a</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Alstom EV17 stage</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sattinger et al. <a href="Zinn">93</a>, 2000</td>
<td>Methane</td>
<td>Magnetostrictive</td>
<td>400 [Hz]</td>
<td>-6 [dB] at 230 [Hz]</td>
</tr>
<tr>
<td>No emissions measured</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>High pressure drop across valve</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evesque et al. [23] (Dowling) 2004</td>
<td>Ethylene</td>
<td>Servo (MOOG)</td>
<td>350 [Hz]</td>
<td>not stated</td>
</tr>
<tr>
<td>Atmospheric test rig of a</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rolls-Royce RB211-DLE GT stage</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Riley et al. <a href="Dowling">90</a> 2004</td>
<td>Ethylene</td>
<td>Servo (MOOG)</td>
<td>350 [Hz]</td>
<td>-30 [dB] at 203 [Hz]</td>
</tr>
<tr>
<td>Similar rig like Evesque et al. [23]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Auer et al.[4] (Sattelmayer), 2005</td>
<td>CNG</td>
<td>Servo (MOOG)</td>
<td>450 [Hz]</td>
<td>-</td>
</tr>
<tr>
<td>No results of ACC-tests published yet</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2.2. Summary of recent literature of active control systems using a modulated gas flow to control instabilities in GT and GT-like combustors
3 Research Approach

In this chapter, the general setup of the atmospheric test facility together with some figures of merit of the apparatus used for most of the operating conditions will be presented. Those special experimental configurations concerning individual experiments will be introduced in the corresponding chapters.

3.1. Test facility

3.1.1. The atmospheric gas turbine combustor test facility

The atmospheric test facility used in the current work is shown in Figure 3.1. It consisted of a single swirl burner (ALSTOM EV5) of 30 to 75 kW thermal power reacting upwardly in a lean premixed mode. The experimental swirl-inducing burner consisted of two slightly shifted half-cones with an estimated swirl number of 0.7 for the operating conditions in the current work. Under the effect of swirling flow, a central recirculation zone was established in the wake-like region stabilizing the flame close to the burner exit (a form of vortex breakdown). In addition, a second concentric recirculation zone was formed downward of the backward-facing step. The Reynolds number based on the average velocity at the burner exit, the burner diameter and the temperature-corrected viscosity of air ranged from 27000 to 42000.

The natural gas and heated air mixed upstream of a flow conditioning section fitted with honeycombs and fine mesh screens to reduce the scale of the free stream turbulence. A plenum chamber was located upstream of the burner. The water-cooled combustion chamber, which area was 1.3 time larger than the upstream duct, had one large quartz window for direct observation of the flame and two small side windows enabling a good optical access for laser based diagnostics. Finally, the exhaust duct was placed downstream of the open end of the combustion chamber in order to ensure atmospheric pressure in the combustion chamber.

Both, the section upstream of the burner and the combustion chamber were equipped with free field condenser microphones for acoustic measurements in the unreactive and
1 Exhaust gas pipe
2 Thermocouple
3 Adjustable temperature controller
4 Exhaust gas cooler
5 Gathering hood
6 Gas detectors
7 Main control unit of the test rig
8 Host computer
9 Target computer
10 Wiring connector block
11 Free field 1/4" microphone
12 Loudspeaker
13 Thermocouple
14 Open access
15 Cooling water unit
16 Cooling jacket of the combustion chamber
17 Tempered holder for a microphone
18 Biaxial optical access
19 Ignition torch
20 Gas injector
21 ALSTOM EV5 burner
22 Flashback arrester
23 Flow straightener
24 Flow meter and pressure reduction valve
25 Air-gas mixer
26 Flow meter and Flow Controller
27 Magnetic safety valve
28 2-stage natural gas regulator
29 Compressed natural gas (CNG)
30 Nitrogen
31 Water temperature controller unit
32 Temperature controller for the air heather
33 Air heather
34 Magnetic valve
35 Flow meter
36 Swing type check valve
37 Air blower & fan

Figure 3.1. Experimental configuration of the gas turbine combustor test rig
reactive flows. In addition, the upstream section of the burner and the combustion chamber were equipped with loudspeakers for acoustic excitation of the flow necessary for acoustic transfer function measurements. Finally, a convergent quartz lens, a narrow pass-band filter (centred at 310[nm]) and a photomultiplier (Hamamatsu, H5783-03) were aligned in front of the quartz window to record OH-chemiluminescence of the flame on the axis of symmetry one diameter downstream of the burner. A more specified characterization of the optical measurement setup is given in Chapter 3.4.

Though the test rig can be fired with different sorts of gaseous fuels, the fuel used within this work has been restricted to compressed natural gas (CNG). Table 3.1 lists the analyzed chemical composition of the utilized natural gas. The regulator (Sherex), reducing the gas pressure, is water heated, capable of accepting varying pressures of the CNG gas bottles (250 – 20[bar]) and of reducing the gas pressure to the working pressure (1.48[bar]) of the mass flow controller (Brooks 5853S). The natural gas entered the mixing device at ambient pressure and delivered volume flow rates from 0 – 300[Nl/min]. The air mass flow rate was generated with a double staged side channel compressor (Rietschle, SHH505) coupled to a volume flow meter (Brooks 5863S, 0 – 2500[Nl/min]).

<table>
<thead>
<tr>
<th>Element</th>
<th>mean</th>
<th>lower limit</th>
<th>upper limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon dioxide</td>
<td>Vol. - %</td>
<td>1.136</td>
<td>0.90</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>Vol. - %</td>
<td>1.763</td>
<td>1.73</td>
</tr>
<tr>
<td>Methane</td>
<td>Vol. - %</td>
<td>90.74</td>
<td>89.03</td>
</tr>
<tr>
<td>Ethane</td>
<td>Vol. - %</td>
<td>4.829</td>
<td>3.84</td>
</tr>
<tr>
<td>Propane</td>
<td>Vol. - %</td>
<td>1.096</td>
<td>0.99</td>
</tr>
<tr>
<td>i-Butane</td>
<td>Vol. - %</td>
<td>0.194</td>
<td>0.13</td>
</tr>
<tr>
<td>n-Butane</td>
<td>Vol. - %</td>
<td>0.304</td>
<td>0.17</td>
</tr>
<tr>
<td>i-Pentane</td>
<td>Vol. - %</td>
<td>0.04</td>
<td>-</td>
</tr>
<tr>
<td>n-Pentane</td>
<td>Vol. - %</td>
<td>0.04</td>
<td>-</td>
</tr>
<tr>
<td>Hexane</td>
<td>Vol. - %</td>
<td>0.05</td>
<td>-</td>
</tr>
<tr>
<td>Total sulfur</td>
<td>mg/Nm³</td>
<td>1</td>
<td>0.1</td>
</tr>
<tr>
<td>Oxygen, hydrogen, helium and water</td>
<td>Vol. - %</td>
<td>0</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 3.1. Chemical composition of the compressed natural gas (CNG) and measured limits of the gas composition during the measurement period (Jan. 2003- Mar. 2005)
The mixture temperature, produced by an electrical air-heater (Leister 10000HT, 15[kW]) has been controlled by a digital PID controller combined with a thermocouple placed in the free stream a few centimetres upstream of the swirl burner.

Air and fuel flow rates, temperatures at various locations and static pressure signals were obtained with a fast multiplexed data acquisition device (NI PCI 6071E) and all set-points of the control system of the test rig were set with a current and voltage output device (NI PCI 6704). These two devices were integrated in a computer running with Windows XP and the data acquisition along with the control program of the test bed has been implemented in Matlab (Matlab R13, Data Acquisition Toolbox 2.6). A detached measuring device was assembled with a simultaneous sample and hold board with eight differential channels (UEI PD2-MFS-8-2M/14). A resolution of 14 bit and (for this device) a rather low sampling frequency of 30[kHz] per acquisition channel ensured sufficient time for the instrumentation amplifier to settle. This configuration allowed an accurate data acquisition with a negligible cross-talk and was primarily used to measure dynamic pressure signals (microphones).

A probe head, placed at the open end of the combustion chamber was used to take a sample of the exhaust gases. Emission components such as CO, CO₂, UHC, NOₓ and O₂ were measured with a Pierburg AMA1000 emissions analysis system (Table 3.2).

<table>
<thead>
<tr>
<th>Species</th>
<th>Type</th>
<th>Measuring principle</th>
<th>Relative error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon monoxide CO</td>
<td>Rosemount,</td>
<td>dry</td>
<td>2%-RS value</td>
</tr>
<tr>
<td></td>
<td>BINOS 1001</td>
<td>NDIR</td>
<td></td>
</tr>
<tr>
<td>Carbon dioxide CO₂</td>
<td>Rosemount,</td>
<td>dry</td>
<td>2%-RS value</td>
</tr>
<tr>
<td></td>
<td>BINOS 1001</td>
<td>NDIR</td>
<td></td>
</tr>
<tr>
<td>Unburned hydrocarbons UHC</td>
<td>Pierburg,</td>
<td>wet</td>
<td>1%-RS value</td>
</tr>
<tr>
<td></td>
<td>PM-2000</td>
<td>FID</td>
<td></td>
</tr>
<tr>
<td>Nitrogen oxide NOₓ</td>
<td>Pierburg,</td>
<td>wet</td>
<td>2%-RS value</td>
</tr>
<tr>
<td></td>
<td>PM-2000</td>
<td>CLD</td>
<td></td>
</tr>
<tr>
<td>Oxygen O₂</td>
<td>Rosemount,</td>
<td>dry</td>
<td>1%-RS value</td>
</tr>
<tr>
<td></td>
<td>BINOS 1001</td>
<td>MPA</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.2. Pierburg AMA1000 emissions analysis system
3.1.2. High speed fuel actuation

For the control and suppression of the combustion instabilities, a secondary fuel injection strategy has been used to reduce the magnitude of these oscillations. On the basis of the geometry given by the combustion chamber the first longitudinal mode (quarter wave mode) could roughly be estimated within the range of $250 \pm 150 [Hz]$ (For an in-depth frequency analysis of the combustion chamber, consult the chapter “Eigenmode analysis of the combustion chamber” on page 51).

However, in many situations, a feedback control strategy requires a mostly linear frequency response characteristic and a resonant frequency of the actuator several times higher than the characteristic frequency of the focused system (here: the combustor). State of the art servovalves were out of the question to fulfil this basic requirement due to the slow mechanical response. Therefore, a solution with a prototype fuel actuator originally developed for a common-rail diesel fuel injection system has been designed. This injection valve was adapted to gaseous fuels and kindly provided for the current work by MOOG Japan Ltd. [104].

The basic design concept of the valve is a tandem arrayed actuation module consisting of two sets of three giant magnetostrictive rods enclosed by a coil to produce a magnetic field to excite an elongation of the tandem array. The two sets of these rods are coupled serially in a Z-shaped holder to eliminate the thermal drift and to double the actuator stroke to maximum of $50 [\mu m]$ (Figure 3.2).

![Figure 3.2. Design concept (left) of the Giant-Magnetostrictive Actuator (GMA) and an image of the complete valve (right) (1 Rod holder, 2 Output shaft bore, 3 Giant-magnetostrictive material)](image)

In contrary to regular common-rail diesel injectors, the giant-magnetostrictive actuator is capable of continuously modulating the forward and the backward stroke and therefore, of permitting sinusoidal injection profiles. In Chapter 6, the injection strategy is described and transfer functions between the excitation signal and the stroke response and between the excitation and the amplitude of the modulated mass flow rate are
measured. For selected operating conditions only, the modulation of an acetone doped mass flow rate arriving at the reaction zone due to the excitation amplitude of the MOOG valve is additionally shown.

Table 3.3 lists a summary of the technical design specifications of this prototype.

<table>
<thead>
<tr>
<th>Specification</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Working fluid</td>
<td>Natural gas</td>
</tr>
<tr>
<td>Norm pressure across the valve, $\Delta p_{valve, norm}$</td>
<td>2.4 [bar]</td>
</tr>
<tr>
<td>Standard mass flow rate at $m_{valve, norm}$ and full stroke</td>
<td>1 [g/s]</td>
</tr>
<tr>
<td>Modulation (normal closed)</td>
<td>0 – 100%</td>
</tr>
<tr>
<td>Target resonance frequency $f_{res}$</td>
<td>1 – 1.5 [kHz]</td>
</tr>
<tr>
<td>Operating temperature</td>
<td>293 – 473 [K]</td>
</tr>
<tr>
<td>Stroke displacement sensor (details: Chapter A.1.1)</td>
<td>AEC-5503A / PU-03A</td>
</tr>
<tr>
<td>High precision power amplifier (details: Chapter A.1.2)</td>
<td>NF 4505</td>
</tr>
</tbody>
</table>

Table 3.3. Technical specifications of the Giant-Magnetostrictive Actuator (GMA) prototype
3.2. Methodology

The experimental approach consisted in a systematic investigation of the qualitative (visual) and quantitative (acoustic) flame characteristics in the parameter space defined by the air/fuel equivalence ratio $\lambda_{afr}$, the mixture preheat temperature $T_{mix}$ and the air mass flow rate $m_{air}$ for two different lengths of the combustion chamber. The objective of this parametric study was threefold: firstly, it aimed at identifying regions in the $\lambda_{afr}$, $T_{mix}$ and $m_{air}$-phase space of stable and unstable flames; secondly, this investigation provided information such as lean extinction limits and qualitative flame shapes and heights; finally, the frequency content of unstable regimes was analyzed. Measurements in the current work comprised flames with air/fuel equivalence ratios from 1.75 - 3.25, mixture preheat temperatures ranging from 500 to 750[K], total mass flow rates from 30 to 46[g/s] and combustion chamber lengths of 0.375 and 0.750[m].

A typical measurement was first initiated by setting the preheat temperature, the total mass flow rate and the air/fuel equivalence ratio to the previously measured lean extinction limit for that combustion chamber length. The air/fuel equivalence ratio was then continuously decreased down to the rich limit by steps of $\Delta \lambda_{afr} = 0.05[-]$ (the total mass flow rate and the mixture temperature were kept constant). The mixture preheat temperature was then increased (steps of $\Delta T_{mix} = 25[K]$) and the procedure was repeated for the entire range of air/fuel equivalence ratios within the temperature-dependent lean and rich limits. Measurements were finally reconducted for the range of desired mass flow rates (steps of $\Delta m_{air} = 2[g/s]$) and for both combustion chamber length.

When combustor operating conditions were changed, new data were typically recorded several minutes after the new condition was set, in order to let the combustor reach a steady operating temperature. At every operating point located inside of the described parameter mesh, a fast sampled and highly accurate data set consisting of four dynamic pressure signals (microphones) and one photomultiplier signal was logged. In parallel, additional data have been recorded, which were composed of two static pressure probe signals (averaged over the acquisition time), of the air and fuel mass flow rates, of five temperatures measured up- and downstream of the flame, of all analyzed emissions as well as of all set points used to control the test rig. All signals were acquired during 10 seconds for each operating condition. The measurements were tested for hysteresis and the results showed only negligible effects with a good repeatability.
3.3. Measurement techniques

3.3.1. Pressure transducers

Free field condenser microphones

The measuring chain to obtain high level and high frequency measurements of dynamic pressures consists of four free-field ¼-inch microphone (Brüel&Kjær, Type 4939), four Falcon range ¼-inch microphone preamplifiers (Brüel&Kjær, Type 2670) and one 4-channel microphone power supply (Brüel&Kjær, Type 2829). Technical specifications, such as dynamic ranges, free-field frequency responses and polarization etc. of the acoustic equipment are listed in the subsections of the appendix “Dynamic pressure measuring devices” on page 162f. Despite the fact that microphone heads are optimised to withstand rough environments and are capable of working at temperatures up to $423[K]$; it was inevitable to cool the microphone and preamplifier to prevent an overheating. To avoid condensation of combustion products on the sensor while keeping the microphones temperature similar for all measurements, the microphones were permanently maintained at a temperature of $348[K]$ ($75[°C]$).

Furthermore, each microphone was calibrated at its temperature of use with an electronic sound level calibrator providing two distinct sound pressure levels at a calibration frequency of $1[kHz]$ (Brüel&Kjær, Type 4231). Figure 3.3 shows a dimensioned drawing of the atmospheric combustor test rig. Although even more holders exist to attach microphones at the combustion chamber, four locations have been selected as a baseline setup for the current investigation.

Possible variations of this setup are introduced in the text when they appear and, in order to simplify matters, the following abbreviations are introduced for the different microphone positions: MIC1 stands for the microphone number one, located 295[mm] down-
stream of the burner exit, MIC2 for microphone no.2, nearby the burner exit close to
the reaction zone and MIC3 and MIC4 for the microphones placed upstream prior to
the EV5 combustor.

**Static pressure probes**

Two static pressure probes were located up- and downstream of the burner in order to
measure the pressure drop across the combustor and the reaction zone (flame). The
pressure transmitter located close-by the burner (Sensortechnics, BT2005G4A) measured
the absolute pressure of the upstream chamber within a range of 0 – 5[bar]. The second
device (Sensortechnics, 423SC01D-PCB) qualified to measure differential pressures, included two sensor heads positioned before and after the burner. This gauge has been designed for operating pressures between 0 and ±68.94[mbar].

### 3.4. Optical measurement techniques

In order to study the dynamics of a combustion chamber, techniques to visualize com­
bustion processes and their response to oscillating pressure fields, etc. are needed to
evaluate stability characteristics of the combustion system. The visualization methods
used to perform these measurements are far-infrared (FIR) absorption, chemilumines­
cence and Planar Laser-Induced Fluorescence (PLIF) diagnostics. There has been an
extensive amount of work done with non-intrusive imaging in the field of combustion.
Gaydon [30] and Eckbreth [21] within their fundamental books and Daily [17],
Hanson et al. [36] and Kohse-Höinghaus [49] are giving excellent reviews of these
imaging techniques related to combustion applications.

For atoms as well as for molecules exists a characteristic set of energy levels (states). However, for diatomic molecules, energy could be simplified as the sum of different
types of motion of the electrons in the electron sheath of atoms: the first level of energy
arises from the orbiting and spin of electrons (\(E_J\)), a second from the vibration (\(E_v\)) and
a third originates from the rotation (\(E_r\)). The spectroscopic terminology uses its own
symbolism to formalise this total sum of energy of the molecule:

\[
T_{tot} = T_v + G(v) + F(J)
\]  

(3.1)

\(G(v)\) indicates the vibration energy, where \(v = 0, 1, 2, \ldots\) is the vibrational quantum
number and \(F(J)\) stands similar for the rotational energy and \(J\) for the rotational quan­
tum number. The quantum state itself is described with \(\Lambda, S\) and \(\Sigma\):

\(\Lambda\): Axial orbital angular momentum quantum number (values: 0, 1, 2, \ldots and greek
letters instead of arabic numbers are used especially along with simple atoms:
\[0 \rightarrow \Sigma; \ 1 \rightarrow \Pi; \ 2 \rightarrow \Delta; \ 3 \rightarrow \Phi; \ \ldots\);\)

\(S\): Total electron spin quantum number (values: 0, 0.5, 1, 1.5, 2, \ldots)
The molecular energy state as well as the transition from one state to another state can be represented graphically (Figure 3.4). The $\Pi$ and $\Sigma$-lines are plotted energy levels in terms of the electronic potential function. The variable $A$ indicates an excited state, e.g. $A^2\Sigma$, in contrary variable $X$ denotes a ground state. $^4\Sigma$, $^2\Sigma^-$ and the $^4\Pi$-lines are examples of electronic configurations resulting in unstable potentials, called predissociative states. Horizontal lines on each potential are showing vibrational states with their corresponding vibrational quantum number $v = 0, 1, 2, \ldots$, where the superimposed thinner lines mark rotational states (the rotational quantum numbers of $J$ are not shown). In molecules with more than two atoms, the complexity of the electron sheath increases incisively and the structure as well as the spectral properties of the molecules need a characterisation in more detail ([17], [49]).

Furthermore, when a transition arises from a lower electronic state to a higher state, a quantum of radiant energy (a photon) is emitted. This radiative process coming from the photon radiation due to transitions of the electrons between the different states, include absorption, induced emission and spontaneous emissions. The release of energy by a photon is given by

$$E = h \cdot v = \frac{h \cdot c}{\lambda} \quad (3.2)$$

Figure 3.5 summarizes the distinct processes of absorbing and emitting photons. Due to the fact that not all transitions are allowed and selection rules exist, the photon are absorbed and emitted only for specific wavelengths. The emitted light has a narrow, sharp wavelength originating from a single transition of a quantum state at the same (resonant fluorescence) or at a different frequency as the absorbed photon or exhibits a wider broadband radiation if several states are involved. Therefore, the wavelength and intensity of the light provides characteristic information of the emitting species. This process of absorption followed by light emission is commonly named fluorescence.

If a transition from a lower to higher state is promoted by a chemical reaction, the term chemiluminescence is used instead of fluorescence (spontaneous emission). Beside photon absorption and chemical processes, photons might be released by heating the
species, by electron bombardments and molecule collisions. Molecules, raised to higher energy state may dissociate, e.g. if the OH radical is excited to a $^4\Sigma$ state, and return to lower energy state via a non-radiative mechanism, such as collisional quenching or intersystem crossing [36].

![Einstein radiative process of a two state system](image)

**Figure 3.5.** The Einstein radiative process of a two state system[17].

### 3.4.1. Chemiluminescence of the OH radical

The experimental simplicity of flame chemiluminescence has set this optical measurement technique to a standard for combustion related diagnostic [18]. Radicals such as $CH^*$, $C_2^*$, $CO_2^*$ and $OH^*$ are present in the reaction zone and are luminous excited. The $CN^*$ and $NO^*$ radicals also radiate, however their luminosity is orders of magnitude lower, even for rich flames. Figure 3.6 shows the emission spectrum of a premixed methane/air flame for the three spectral bands corresponding to $OH^*$, $CH^*$ and $C_2^*$. The air/fuel equivalence ratio $\lambda_{efr}$ has a strong influence of the intensity of the radiation signals. The intensity of the light emitted by the $CH^*$ and $C_2^*$ radicals is strong in stoichiometric to rich conditions and quite weak to marginal for lean operating points. Unlike these radicals, the emitted radiation from the $OH^*$ radical shows a better characteristic for stoichiometric up to lean equivalence ratios. The emission spectrum of the $OH^*$ radical is based on the transition $A^2\Sigma^+ \rightarrow X^2\Pi(0 - 0)$ with a bandhead wavelength of $308.9[nm]$ $(Q_2)$ and is usually chosen because it is only little dependent on temperature.

The EV5 combustor was solely operated at lean conditions and therefore chemiluminescence of the $OH^*$ radical has been exclusively studied. For this work, chemiluminescence of the OH radical can be categorized into two groups: measurements using a single photosensor module obtaining an spatially averaged signal of the reaction zone and a two dimensional imaging technique.
3.4.1.1. Photosensor module

A photosensor module, including a photomultiplier tube (PMT) and a high voltage power supply, has been used. The luminosity of the flame was collected with a convergent quartz lens and led through an optical narrow pass-band filter centred at \(310\,[\text{nm}]\) following a photosensor module (Hamamatsu H5783-03). The photo current from the photomultiplier tube was monitored as the voltage drops across a 100-kV load resistor, which was lowpass filtered and amplified. (See Chapter A.1.4 for technical specifications).

Compared to imaging techniques, the photosensor module has the drawback to obtain only integrated one-dimensional information. Hence, the advantage of PMTs of delivering a continuous signal of the combustion process makes it possible to compare the chemiluminescence signal with dynamic pressure signals.

3.4.1.2. 2-D chemiluminescence of the OH radical

For two dimensional measurements, the emitted light has been collected onto a Charge Coupled Device (CCD) chip of a camera to acquire a spatially resolved chemiluminescence signal. Because of the low intensity of the emitted radiation, the camera has an intensified CCD array (ICCD). Similar to the photosensor module, every ICCD matrix pixel collects the light emitted by the flame only along the optical path ("line of sight" measurements). On account of this, chemiluminescence measurements cannot capture fine structures or even the situation at different \((x, y)\)-levels \((z\) being the axis of observation) in the flame, since the signal is integrated through the depth of the flame.

![Figure 3.6. Spectral bands for a premixed methane/air flame at \(\lambda_{\text{eff}} = 0.9\) and atmospheric conditions. The spectra is recorded with a CHROMEX grating spectrograph [18].](image-url)
Another disadvantage of chemiluminescence is that the signal strength is several orders of magnitude lower than Laser Induced Fluorescence (LIF). This will decrease the temporal resolution of measurements, since longer integration times are required to obtain sufficient signal strength. Nevertheless, chemiluminescence measurements are much more convenient to apply since they do not require a costly light source (e.g. laser) and they can be acquired not only phase-locked, but rather continuously, if a fast camera is available.

**Optical arrangement**

The intensified CCD camera was aligned in front of the main window (fused silica) of the combustion chamber (See Figure 3.7). The ICCD camera (DiCAM Pro) has a resolution of $1280 \times 1024$ pixel (SVGA), a 12 bit dynamic range and the possibility to change the exposure time between $5\,\mu\text{s}$ and $1000\,\mu\text{s}$. The photocathode S20 has a detection range from $170\,\text{nm}$ up to $680\,\text{nm}$ and a peak wavelength at $430\,\text{nm}$ providing an excellent spectral sensitivity within the UV range.

The fused silica objective is a tailor-made product with a luminous intensity (candle power) of $f/2$ and focal length of $100\,\text{mm}$. A WG305 long pass and a bandpass filter with a center frequency of $310\,\text{nm}$ ($FWHM = 11 \pm 2\,\text{nm}$) were placed on the objective to prevent illumination at other wavelengths not emitted by the $A^2\Sigma^+ \rightarrow X^2\Pi(0-0)$ transition (Q2 bandhead).

---

**Figure 3.7.** Setup for the OH-Chemiluminescence measurements
Optical measurement techniques

Triggering configuration

Since phaseaveraged (phase-locked) measurements were of interest, a phase reference signal and the camera have to be synchronized. Microphone No.2 (MIC2) was used as the reference signal over the entire range of operating conditions. The camera has the ability to read out eight frames per second and therefore, an additional dead time with respect to this $8[Hz]$ has been superimposed on the TTL resulting from the positive zero crossings of the lowpass filtered pressure signal. This configuration prevented the camera from a pre-trigger before the camera read out the data of the previously acquired image.

To take images for every 45 degrees ($T/8$) of the period of the dynamic pressure signal, a time delay corresponding to the desired phase position has been added to the TTL to obtain the final trigger signal $\Delta \Phi_{\text{trig}}$. At all of the 45 degree phase steps, the camera acquired 140 images with an exposure time of $400[\mu s]$. The relative timing needed for the synchronization was implemented with a Stanford DG535, 4-channel digital delay/pulse generator.

![Figure 3.8. Triggering configuration for the chemiluminescence measurements](image)

3.4.2. OH-Laser Induced Fluorescence

Planar Laser Induced Fluorescence (PLIF) measurements are able to capture fine structures in the flame, where chemiluminescence measurements cannot. PLIF images are taken on a specific vertical plane where the species of the flame are excited by the laser sheet. Combined with a shorter integration time to obtain an image (temporal resolution), PLIF has an excellent spatial resolution. Laser induced fluorescence is spontaneous emission following an absorption event from a molecular tracer excited by laser
Optical measurement techniques

radiation. The tracer can be naturally occurring (e.g. $OH$, $CH$, $CH_2$, $C_2$, $CN$, $NO$, $NO_2$, $HCO$ and $CH_3O$ in flames) or seeded into the flow (e.g. acetone). A portion of the tracer molecules absorbs the incident photon bringing the atom to an excited state. Is such a state unstable, then another photon is spontaneously emitted with a characteristic wavelength.

In addition, a collision can populate adjacent states resulting in fluorescence from states close to the excited states. This observed LIF spectrum depends not only on the selected tracer but also on the individual line transition probability, the population of the excited quantum states that are emitting photons and the line shapes of the individual transitions. Within this work, mainly the $A^2\Sigma^+ \rightarrow \chi^2\Pi(1-0)$ band of the $OH^*$ radical was excited via the $Q_1(8)$ transition at 283.55[nm]. Therefore, stimulated emission of the $OH^*$ was selected to detect in the (1−1) vibrational band at 312.2[nm].

**Optical arrangement**

To generate the UV beam at an excitation wavelength of 283.55[nm], a frequency doubled, YAG pumped dye laser system was used. The pumping source, a pulsed Nd:YAG laser is a solid state laser (Quantel, YG981A-20) with a wavelength of 532[nm] (fundamental wavelength of 1064[nm]) and operates at rate of 20[Hz] (pulsed energy of 600[mJ]). The standard oscillator of the dye laser (Quantel TDL50; dye: Rhodamine 6G) gives a 0.08[cm$^{-1}$] line width at 560[nm]. Using the UV extension module, a spectral range of 360−420[nm] is tunable with minimal wavelength displacement of 0.25[pm]. At the exit of the dye laser, the UV beam had a diameter of less than 5[mm] and pulse duration shorter than 10[ns].

The dye laser beam was adjusted with two mirrors and lined up with respect to the gas turbine combustor (See Figure 3.9). A lens aperture prevented unintentional reflections and thereafter, the beam was formed to a laser sheet with three lenses achieved by a plan-concave lens (focal length $f = −50[mm]$) spreading the beam in a horizontal plane, by a plan-convex lens ($f = 100[mm]$) focusing the beam into a vertical sheet and finally by a cylindrical lens ($f = −200[mm]$) correcting the divergence of the laser sheet. The thickness of the laser sheet across the measurement section was about 500[μm] at a height of the sheet of 80[mm]. To determine the absorption, the shot-to-shot variations of the laser intensity and the laser sheet inhomogeneity, the profile of the laser sheet has been recorded before and after the measurement section with an additional CDD camera (PCO Sensicam, 12bit, 1280 x 1024 SVGA). 95% of the light sheet passed through a beam splitter and entered the reaction zone by a smaller fused silica window, crossed the flame stabilization zone and left the reaction zone by an additional window. This laser sheet and the remaining 5% of the sheet reflected by the beam splitter have been redirected to a separate screen, where the CCD camera was focused on.
1 Triggering unit & PC  
2 Nd-Yag laser  
3 Dye laser  
4 Sheet forming lenses  
5 Beam splitter  
6 UV mirror  
7 UV main window  
8 Combustion chamber  
9 Microphone  
10 Glass screen  
11 CCD camera  
12 Protection shield  
13 ICCD camera  
14 UV objective  
15 Passband filter  

Figure 3.9. Setup for the OH-Laser Induced Fluorescence measurements

The imaging of the reaction zone with the ICCD camera, perpendicular the laser sheet, and the UV objective were identical to those in the chemiluminescence measurements.

**Triggering configuration**

Similar to the chemiluminescence triggering setup, the signal from MIC2 was applied as a reference and a TTL derived from its zero crossings. The Nd:YAG laser showed the best performance (stability, intensity) at the pulse repetition rate of 20[Hz]. Therefore, an additional time delay has been superimposed on the TTL. After the TTL showed a raising edge and a trigger occurred, a time delay $t_{phase}$ corresponding to the desired phase angle has been added. In addition, a dead time of 0.049[s] has been added to prevent a synchronization of the laser system with a frequency higher than 20[Hz] (no pre-triggering allowed). The resulting signal $\Delta \Phi_{trig}$ was used to trigger the flash lamps of the laser Nd:YAG laser. The time delay used by the response of the flash lamps, the Q-switch and the time till radiation is emitted by the laser was in the order of magnitude of 580[\mu s]. This small time delay misleadingly seems to be negligible. But compared to the pulse length of the laser of 20[ns] and the gate width of the cameras, an additional time delay $t_{delay}$ was needed to synchronize both of the cameras to the laser pulse.
The gate width of the CDD and the ICCD camera was chosen and minimized to keep the iris open only during the laser pulse and the resulting fluorescence of the $OH^*$ radical. Therefore, the ICCD camera was triggered 180[ns] later than the laser system with a gate width of 60[ns]. In contrary, the CDD camera imaging the laser sheet profiles was used with a gate width of 1[ms] and according to this time scale, no additional time delay was needed to gather the corresponding laser pulse. Every third laser pulse has been recorded with the cameras since the frame rate to read data out was not faster than 8 frames per second.

![Figure 3.10. Triggering configuration for the LIF measurements](image_url)
3.4.3. Acetone-Laser Induced Fluorescence

In many studies, acetone has successfully been adopted as a tracer for PLIF [98] since acetone fluorescence has a linear characteristic with species concentration and laser power (for isothermal and isobaric flows). Lozano et al. [60] carried out a detailed investigation of the photophysical properties of acetone and showed the possibility to measure concentration rates with acetone. Thurber and Hanson [106] extended this work and explored pressure, temperature and composition effects for three excitation wavelengths and improved the understanding of the temperature and excitation wavelength dependencies of the fluorescence of acetone.

Acetone (dimethyl ketone, \( CH_3COCH_3 \)), liquid at room temperature, has a high vapour pressure and a low toxicity. These basic properties permit a simple seeding strategy and make it possible to seed a gaseous flow at high concentration rates. Furthermore, an accessible absorption in the broad spectral range of 225 – 320[nm] (max. at 280[nm]) and fluorescence within the range of 350 – 550[nm] features allow an excellent application to PLIF.

Acetone fluorescence measurements have been performed in a similar configuration of the combustor test rig as previously used along with the \( OH^* \) PLIF technique. In addition, two measurement setups were built beside this standard configuration. The first arrangement (Figure 3.11; left) was set by integrating a second optical unit to obtain an insight into the upstream section of the combustion chamber prior to the EV combustor. Acetone was seeded into the natural gas mass flow by bubbling into two serially arranged baths filled with liquid acetone. The two baths were temperature stabilized at 293[K] and continuously monitored. Assuming a constant vapour pressure at the top of the final bath equal to the saturation pressure, the concentration of acetone was only influenced by the natural gas flow rates through the baths. The second, additional configuration (Figure 3.11; right) beside the standard setup was used to determine the range of frequencies where modulated injection rates of the high speed fuel actuator are possible. The gas flow through the valve was therefore seeded with acetone and continuously injected as a modulated free jet into ambient conditions.

Optical arrangement

The laser system, including the excitation wavelength, the laser sheet generation, the detection devices (cameras and objectives) and the triggering scheme, was used in a similar manner as described in Chapter 3.4.2. The fluorescence of acetone was collected with the previously described ICCD camera with two different filters in front in order to prevent penetration of wavelengths outside the desired fluorescence spectrum of acetone (shortpass filter: Schott BG39, longpass filter: Schott WG335).
3.4.4. Methane absorption measurement technique

Laser absorption is one of the most common optical techniques due to its simple instrumentation and data evaluation ([20],[33],[49] and [54]). Radicals detectable with an adequate sensitivity and accuracy include $NCO$, $NH$, $CH$, $NH_2$, $CN$ and $CH_3$, where this work focuses on the absorption of the $C-H$ stretching band of hydrocarbons (mainly those of $CH_4$). Beer's law describes the decrease of the intensity due to absorption of monochromatic light:

$$I = I_0 \cdot e^{-(k_x l c_x)}$$  \hspace{1cm} (3.3)

The incident intensity $I_0$ is exponentially reduced along the laser beam path $l$ to the resulting intensity $i$ at the detector. $k_x$ is the frequency-dependent absorption coefficient of the species $x$, whereas $c_x$ stands for the concentration. Beer's law indicates a severe drawback of absorption techniques based on the fact that a spatial resolution of concentrations along the beam is rather impossible without specific assumptions due to the line of sight character of this measurement technique. In the near infrared spectrum, the transition of the $v_3(P7)$ line of methane has been determined by high resolution infrared absorption spectroscopy and demonstrated an excellent capability to absorb the 3.39[$\mu m$] $He-Ne$ laser line by showing a weak dependency on temperature variations up to 1500[K] [61].

Optical arrangement

The incident wavelength was generated with a cylindrical helium neon laser head (Research Electro-Optics, Inc.; REO LHIP-0201-339) emitting infrared radiation at a
wavelength of $3.39[\mu m]$ with $2.0[mW]$. The diameter of the beam at the output of the laser was $2.03[mm]$ and the beam divergence specified by the manufacturer is given with $2.13[mrad]$. To monitor the laser power variations, an indium arsenide infrared photovoltaic detector has been used. This thermoelectrical cooled sensor (PVI-2TE-InAs) has a response time smaller than $2[ns]$ and was mounted in housing combined with a preamplifier and a Peltier cooling element (VIGO system, DR-1B-CTTC-02/230).

Three different experimental configurations came into consideration:

- The laser was mounted at the test rig in front of the small window at the right side of the combustor and the beam was directed to the small window at the left side, whereby the beam crossed the exit of the EV5 burner (separation distance between beam and burner exit $<2[mm]$). The free optical path length between the laser and the InAs detector was $200[mm]$ and the absorption path length was commensurate with the diameter of the combustion chamber ($94[mm]$).

- An identical setup as aforementioned has been established at the upstream section of the test rig to measure equivalence ratio fluctuations in the premix section (cp. Figure 3.11, left).

- To verify injection rates of the high speed fuel actuator, absorption measurements were performed at the outlet of the valve (Figure 3.11, right). The laser and the detector were aligned across the center of the outflow at an axial distance of $3[mm]$. The absorption path length had the order of magnitude of the diameter of the exit tube ($3[mm]$) and the distance between laser and detector was set to $100[mm]$.

### 3.5. Acoustic signal processing and analysis

A regular measurement of the dynamic pressure signals consisted of 10 second-long data acquisition phase. Thereafter, the signal recorded as a voltage has been multiplied by a conversion factor to obtain a pressure value (Pascal) in the time domain. In a succeeding step, a harmonic analysis was performed to achieve a frequency domain characterisation of pressure signals. Therefore, signals were divided into slices of 2 seconds, with an overlap of 75% ensuring statistically independent spectra and in order to obtain a desired effective bandwidth of $1[Hz]$ while maintaining a good signal-to-noise ratio, a minimum 4-terms Blackman-Harris window was applied on each slice.

Finally, a Fast Fourier transform (FFT) was performed along each window. Hence, the resulting pressure spectrum is an average of this individual and independent spectras. This procedure has been previously applied to purely sinusoidal samples of known frequency and amplitude. Thus, a correction factor inherent to the method could be
achieved to correct the Sound Pressure Levels (SPL) in the frequency domain. Furthermore, a filter method has been implemented to remove the “noise floor” of the pressure spectra to analyse and compare the various dominant modes present in the gas turbine combustor.

The following sections provide an insight into the acoustic signal processing approach used within this work. Basic definitions of acoustic properties, such as those of the decibel scale, of the sound pressure level, of the sinusoidal wave etc., are given in the Appendix “Additional theory of acoustics” on page 165 et seq.

3.5.1. Calibration of the sound pressure level (SPL) of the microphones

Standard dB levels, emitted by a sound level calibrator operating at $1[kHz]$ are independent from the use of a weighting network (A, B or C weighting) as a result of the unity gain defined for all weightings at response frequency of $1[kHz]$. Two distinct levels, $94[dB]$ and $114[dB]$ for noisy environments are freely selectable. Using the definition of sound pressure levels along with the reference pressure $p_{ref} = 20\mu Pa$ and the calibrator level of $114[dB]$, the root mean square value of the physical pressure signal in Pascal [Pa] can be derived. Since a difference between the two levels of $20[dB]$ exists, the $94[dB]$ level has a ten time smaller value represented in Pascal than the $114[dB]$ level.

$$p_{cal,114, rms} = p_{ref} \cdot 10^{\frac{114 dB}{20 dB}} = 10.0237 Pa$$ (3.4)

At this point, the calibration of microphones is straightforward. A reference measurement with the $114[dB]$ level of the calibrator and the microphone is performed. The data acquisition system, the sampling rate and the sampling period, the amplifier gain and the specifications of input filters influence the calibration factor due to a changing RMS value and these values kept therefore constant for all measurements. For simplicity, the conversion factor $c_{V_{io}Pa}$ calculated from the RMS value of the voltage signal $V_{rms}$ and from the RMS value of the pressure $p_{cal, rms}$ is defined with the units of $[Pa/V]$. Thus, for a signal generated with the acoustic calibrator at a level of $114[dB]$, the conversion factor $c_{V_{io}Pa}$ has the following appearance:

$$c_{V_{io}Pa,114 dB} = \frac{p_{cal, rms}}{V_{meas, rms}} = \frac{p_{ref} \cdot 10^{\frac{114 dB}{20 dB}}}{V_{meas, rms}} = \frac{10.0237 Pa}{V_{meas, rms}}$$ (3.5)

Depending on temperature, moisture, ageing and input impedance of the data acquisition system, this factor changes significantly, e.g new microphones have a conversion
Acoustic signal processing and analysis

factor of 250 [Pa/V] stated by the manufacturer. Calibration factors of the actual microphones after 10 and 125 operating hours in the reactive flow can be seen in Table 3.4.

<table>
<thead>
<tr>
<th></th>
<th>10 operating hours</th>
<th>125 operating hours</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$c_{VtoPa,94dB}$</td>
<td>$c_{VtoPa,114dB}$</td>
</tr>
<tr>
<td>MIC1 [Pa/V]</td>
<td>816.16</td>
<td>1241.2</td>
</tr>
<tr>
<td>MIC2 [Pa/V]</td>
<td>731.33</td>
<td>1299.2</td>
</tr>
<tr>
<td>MIC3 [Pa/V]</td>
<td>777.93</td>
<td>1355.1</td>
</tr>
<tr>
<td>MIC4 [Pa/V]</td>
<td>307.61</td>
<td>596.24</td>
</tr>
</tbody>
</table>

Table 3.4. Calibration factors $c_{VtoPa,94dB}$ and $c_{VtoPa,114dB}$

3.5.2. Harmonic analysis (Fourier Transform / FFT)

To estimate the frequency content of a function, the Fourier Transform is a fundamental mathematical method to map the time domain onto the frequency domain. The function $f(t)$, assumed to be periodic, can be represented by an infinite sum of sine and cosine functions, called Fourier series. For a function periodic in $[-T/2, T/2]$, the series can be written as

$$A_n = \frac{1}{T} \int_{-T/2}^{T/2} f(t) e^{-2\pi i n t / T} dt$$

(3.6)

where $A_n$ denoting the coefficients for the Fourier series expansion. In addition, replacing the discrete coefficients $A_n$ by its continuous representation while letting $T \to \infty$ ($n/T \to \nu$) and using the angular frequency $\omega = 2\pi \nu$, the generalized formulation of the continuous Fourier Transform can be derived (Equation 3.7).

$$F(\omega) = \int_{-\infty}^{\infty} f(t) e^{-j\omega t} dt, \text{ where } |F(\omega)| = 0 \text{ and } |\omega| \geq \frac{1}{2}(2\pi/T)$$

(3.7)

However, signals acquired by a data acquisition system are of finite extent and the integration of the Fourier Transform can only be performed by a finite summation of finite samples. Considering signals and not functions, the infinite integration of equation 3.7 has to be replaced by the summation over the $N$ samples of the signal.

$$F_n(\omega) = \sum_{n=-N/2}^{N/2} f(nT) e^{-j\omega n T}$$

(3.8)
In particular, the start point of the "sampled" Fourier Transform $F_s(\omega)$ can be shifted by $N/2$ without any loss of information (only by affecting the phase angles of the transform). The resulting transform $F_D(\omega_k)$ is often called the forward Discrete Fourier Transform (DFT).

$$F_D(\omega_k) = \sum_{n=0}^{N-1} f(nT)e^{-j\omega_k nT} \quad \text{where} \quad \omega_k = \frac{2\pi}{NT} k \quad \text{with} \quad k = 0, 1, 2, \ldots, N-1 \quad (3.9)$$

Calculating the DFT contains a lot of redundancy, therefore all efficient computational programs use the fundamental algorithm first presented by Cooley and Tukey [15] to fasten their computations. This technique is known as the Fast Fourier Transform FFT. The assumptions made by the change from the continuous to the discrete time space (from equation 3.7 to 3.8) have undesired effects. Frequencies of the signal periodic to the signal length are transformed, whereas all other frequencies exhibit zero protections. This spectral leakage can be reduced by special weighting functions (windowing) by removing any discontinuity at the two endings of the finite data record.

Another loss is introduced to the finite frequency resolution of the spectra (picket-fence effect). The frequency resolution of the DFT is given by the sample rate divided by the total number of samples of the signal. All frequencies not resolved by the discrete spectrum are binned (melt) at the next resolved frequency. Therefore, an error in both frequency and amplitude is located. Thus, the uncertainty of frequency is smaller than the frequency resolution of the DFT. In contrary to the frequency, the error of the amplitude can be influenced by the use of a weighting function ([29],[82]).

The spectral leakage and the picket-fence effect are minimized according to the work of Harris [42] by adapting a 4-term Blackman-Harris weighting function to the time domain of the pressure signal. This window has a highly concentrated central lobe with a fast sidelobe falloff and the overlap correlation (by 75% overlapping) is smaller than 46%. To prevent loss of information, the 10 second-long signals are divided into 15 sequences by a 75% -overlap and multiplied with the weighting function.

Afterwards, the sequence of the acquired samples has been zero padded to obtain a vector of length $n^2$. (If the number of samples is a power of 2, the FFT algorithm has its best efficiency and the computation time is especially reduced). In a next step, the FFT has been calculated for every zero-padded and windowed sequence. At the end, the 15 spectra resulting from the sequences have been average to final spectra with a frequency resolution $\leq 1[Hz]$. 
3.5.2.1. Scaling of the levels of the Fast Fourier Transform (FFT)

The spectral estimate of a signal is influenced by the usage of a weighting function. For instance, a 4-term Blackman-Harris window reduces the signal to zero values near the boundaries. This processing loss reduces the energy of the signal and lowers the level of the spectra. A reverse effect, a processing gain is supposable as well. For the use in the current investigation, a proportionality factor has been experimentally derived to scale the bias on spectral amplitudes inherent to the signal processing. To this end, a signal emitted by the sound level calibrator has been recorded, pressure calibrated according to section 3.5.1 and transformed from the time to the frequency domain with the previously defined method. The \( 114\text{[dB]} \) level defined for the calibrator (at \( 1[kHz] \)) has been compared with the actual value of the spectral amplitude at \( 1[kHz] \) and a calibration factor calculated (The factor itself is applied to the time domain).

3.5.2.2. Filtering a spectrum and SNIP

By filtering a spectrum, the used method is qualitatively well adapted if the higher moments of the peak and its linewidth are preserved and if the resulting curve is visibly smoothed. Such a well-adapted filter for line smoothing of a spectrum is the Savitzky-Golay filtering method. The filter can be described as a generalized moving average filter, where the function of the moving window is more generalized. Instead of only averaging values along the predefined window, the underlying function calculates a higher order polynomial least-squares fit of the data points matching the window. At the next point of the spectrum, a new fit has been calculated within a shifted window.

Furthermore, to estimate the noise floor of spectra and to compare dominant modes, a more inconvenient characteristic of a filter is desirable. A background treatment of X-ray spectra used by the analysis of geoscience data allows to separate continues background information from peaks. The Statistic-sensitive Non-linear Iterative Peak-clipping (SNIP) algorithm presented by Ryan et al. [91] demonstrated a good performance. The high frequency content of the signal is smoothed, higher peaks clipped and the lower level of the signal remains unaltered.

\[
f_{\text{SNIP}}(x) = \min\{f(x), [(f(x + i) + f(x - i))/2]\}
\]  

(3.10)

The filtered sample \( f_{\text{SNIP}}(x) \) at position \( x \) is the lesser of the value of the origin sample \( f(x) \) or of the mean value of the two samples located at a distance of \( +i \) and \( -i \) away from \( f(x) \) (2\( i \) is the filter scanning width).
3.6. Digital image post-processing and analysis

3.6.1. Image correction methodology

Digital imaging, quantitative as well as qualitative, requires always an appropriate image correction. The optical arrangement, the response of the camera, the laser sheet intensity distribution, the laser shot-to-shot intensity variations, the absorption along the laser beam and the influence of the background were considered as possible error sources.

By a first step, the actual response of the ICCD camera \( I(i,j) \) needs to be calibrated to account for non-uniform pixel characteristics. To this end, an image has been collected of a uniform illuminated field \( I_{IP}(i,j) \) (e.g. a twilight flat is an easy obtainable illuminated field by imaging a blank spot in the sky at twilight.) and subtracted from a dark image \( I_{DF}(i,j) \) recorded with no illumination present (with the cap of the lens on the objective). The resulting image \( I(i,j) \) can be calculated as

\[
I(i,j) = \frac{I(i,j) - I_{DF}(i,j)}{I_{IP}(i,j) - I_{DF}(i,j)}
\]  

(3.11)

Furthermore, the fluorescence signal of the instantaneous image \( I(i,j) \) at the pixel location \((i,j)\) is subtracted by an averaged intensity image of the background \( I_{f,BG}(i,j) \) acquired with a running experiment, but with no laser excitation present at the measurement section. This subtracts any influence of the ambient light and any fluorescence already present in the experiment.

\[
I_{f,backcorr}(i,j) = [I(i,j) - I_{f,BG}(i,j)]
\]  

(3.12)

In a third place, the images taken with the second CCD camera recording the two laser beams came into consideration (See Figure 3.9). The images were corrected according to equation 3.12 (and 3.11) and split into two regions of interest (ROI) showing two vertical fluorescence stripes: the first region shows the beam directly stemmed from the beam splitter \( ROI_{LS,in} \) and the second region \( ROI_{LS,out} \) displays the beam after the combustion chamber. The 140 images recorded at the beginning of each measurement were used to determine reference scale values. The reference scale value for the shot-to-shot intensity variations of the laser sheet \( R_{ROI_{LS,in}} \) has been derived by calculating the mean intensity for the region before the combustion chamber \( ROI_{LS,in} \) by summing up all the rows and columns of this region and by averaging this mean intensity out of the 140 images. Hence, the calibration factor for the shot-to-shot intensity variations of the laser sheet \( C_{sts} \) can be derived by computing the average of the first region for every
actual image $I_{ROI, LSin}$ during the measurement and by dividing its reference value. Hence, the correction of varying intensity is made on a single shot basis.

$$C_{sts} = \frac{I_{ROI, LSin}}{R_{ROI, LSin}}$$  \hspace{1cm} (3.13)

The average of the second specified region $R_{ROI, LSout}$ is similarly calculated. The ratio of the regions $R_{ROI, LSout}$ and $R_{ROI, LSin}$ is representative for transmission losses in relation to the windows and losses due to imperfect reflections at the mirrors. Taking into account the ratios calculated by the actual images, an intensity difference originating from the absorption of the laser beam by the tracer is computable.

$$\Delta I_{abso} = I_{ROI, LSin} \cdot \frac{R_{ROI, LSout}}{R_{ROI, LSin}} - I_{ROI, LSout}$$  \hspace{1cm} (3.14)

Assuming a constant concentration of the tracer within the measurement section, Beer’s law (Equation 3.3) of absorption can be rearranged and the factor $c_xk_x$ enumerated.

$$I = I_0 \cdot e^{-(c_xk_x)} \Rightarrow \langle c_xk_x \rangle = \frac{1}{I_0} \cdot \ln\left(\frac{I_0}{I}\right) = \frac{1}{I} \cdot \ln\left(\frac{I_{ROI, LSout} + \Delta I_{abso}}{I_{ROI, LSout}}\right)$$  \hspace{1cm} (3.15)

In a next step, a matrix $(C_{Abso}(i,j))$ is built to correct every single pixel of the actual image and adjust the influence of absorption on the actual image. The length $l(i,j)$ from the incident beam to the pixel at the position $(i,j)$ includes the diverging nature of the laser sheet.

$$C_{Abso}(i,j) = l(i,j) \cdot e^{\langle c_xk_x \rangle}$$  \hspace{1cm} (3.16)

And finally, the laser sheet inhomogeneity contributes to stripes in the resulting image and therefore the distribution along the vertical direction of the laser sheet is obtained to correct this influence. The pixels of the two regions of the camera used for the corrections $(ROI, LSin$ and $ROI, LSout)$ are horizontally summed resulting in two 1-row binning images representing the intensity distribution of the laser sheet. The method presented by Seitzman et al. [99] has been used to estimate the geometric expansion and the difference in magnification between the CDD and ICCD camera. The incident laser sheet was partially blocked at the upper and lower range of the sheet and recorded with the CCD camera. At the same time, the emitted light of the OH radical was acquired with the ICCD camera.

The partially blocked laser sheet produces stripes with no intensity on the OH image. A correlation between the left part of the OH image and the 1-row binning image of $ROI, LSout$ and similar behaviour with $ROI, LSin$ derives all required geometric prop-
erties to correct the images. The intermediate rows between the obstacles were linearly interpolated. With this information, a correction matrix $C_S(i,j)$ has been calculated on shot-to-shot basis to correct the stripes of the OH image.

Every single image has been corrected with the following set of equations:

$$I_c(i,j) = \frac{I(i,j) - I_D(i,j)}{I_{IL}(i,j) - I_D(i,j)}$$

$$I_{corr}(i,j) = [I(i,j) - I_{f, BG}(i,j)] \cdot C_{S\&S} \cdot C_{ABG}(i,j) \cdot C_S(i,j)$$

A final transform, after the application of the correction procedures, has been applied to correct geometric distortions and represent the pixel on a metric scale. A correction image has been recorded which shows a rectangular grid placed in the test section. Several points at the crossing of two gridlines (with a predefined distance from each other) were used as reference points to interpolate and remap the image.

3.6.2. Further image post processing

Flame front detection

The OH radical is present in hot sections of the combustion gas as well as in the flame front itself. Therefore, it is possible to apply a rather simple procedure since the values of the intensity of the OH images have a fast transition at the flame edges. Pixels with higher intensity tend towards the flame edge, whereas pixels with lower intensity rather correspond to the area of the burned gas ([34],[37],[55]). To estimate a proper threshold value to identify the flame front, a PDF of the fluorescence intensity was calculated.

Figure 3.12. Unscaled OH LIF image without corrections (left) and a scaled OH LIF measurement including image corrections (right).
and lowpass filtered for every image. A typical bimodal distribution is shown in Figure 3.13. The threshold value $\hat{2}$ to identify the flame front has been estimated as the inflection point located between the minima $\hat{1}$ and the second maxima $\hat{3}$ of the bimodal distribution.

All pixels having a higher or at least the same intensity of the threshold were set to one (red), all the others to zero (black). The resulting binary image is cut into two areas along the flame front showing a burnt and unburnt zone whereby the flame front is defined by borderline between the two zones. To reduce the noise along the flame front resulting from digitizing the image, the coordinates $(i, j)$ have been filtered with a low pass filter.

The combustion probability is worked out by averaging several of these binary images at a particular operating condition. In addition, the probability of the localization of the flame front is derived similarly by averaging multiple flame front images. Figure 3.14 shows OH LIF single shot image and the resultant post-processed images.

### Center of gravity

A method according to Assen et al. [5] was used to estimate the object center of an image. The so-called center of gravity method ($CoG$) was applied to intensity images. This allows an identification of the regions with concentrations where the LIF tracer is intensely present. Equation 3.18 defines the use of the $CoG$-method.

$$CoG(i, j) = \frac{\sum_{i,j} i \cdot w(i, j) \sum_{i,j} j \cdot w(i, j)}{\sum_{i,j} w(i, j) \sum_{i,j} w(i, j)}$$  \hspace{1cm} (3.18)
where \( w(i, j) \) is a function that weights the coordinates according to \( w(i, j) = m - I(i, j) \) by applying \( m = \max(I(i, j)) - 0.01 \). In this way, the center of gravity is attracted by pixel with a higher intensity.

**Region of interest**

The mean intensity values and the center of gravity was calculated within several distinct regions of the corrected images. The main region of interest (ROI0) consists of the entire range of the image whereby needless information, e.g. pixels of the border of the window, etc. has been neglected (See Figure 3.15). The second and the third regions of interest (ROI1-left side, ROI2-right side) as well as the region of interest ROI3 (center part) had the same amount of pixels and showing individual sections (left, right and center) at a similar size (See Figure 3.15).
Figure 3.15. Regions of interest (ROI0 - ROI3).
4 Fundamental Characterization

4.1. Eigenmode analysis of the combustion chamber

The coupling between unsteady heat release and acoustic waves is fundamental for combustion driven oscillations. The geometric enclosure of the combustor acts as a resonant system and at the boundaries of the burner, pressure waves are reflected. This reflected pressure waves may interact with the heat release process. If the dissipation of acoustic energy within the combustion chamber and at the boundaries is smaller than the energy gain of the acoustic disturbances yield by the combustion process, pressure waves are excited and pressure amplitudes will grow up to a saturation limit (the limit cycle). Since the acoustic response is an integral part of self-excited combustion oscillations in confined flows, the resonance frequencies of the combustor geometry play a decisive role. In order to derive the acoustic characteristics of the geometry for the present combustion system, two simple approaches were made to calculate resonance frequencies.

An elementary method to estimate resonance frequencies of this type of combustor implies the following rough approach. The responsible part of the combustion chamber for the acoustic modes consists only of the cylindrical downstream section. All other geometries of the gas turbine combustor are neglected and neither a reaction zone nor a mass flow rate are considered to have an influence on acoustic properties. The boundary conditions defined at the walls of the cylinder are hard sound boundaries. The same boundary was used for the position of the EV burner. At the open end of the duct at the exit of the combustion chamber a pressure node is located, where on the other end at the burner, an antinode exists.

The first resonant frequency of this system at the fundamental quarter wave mode and all the higher orders of the longitudinal mode \( f_i \) depend only on the speed of sound \( c \) and the length of the duct \( L \).

\[
f_i = l \cdot \frac{c}{4L}, \text{ where } l = 1, 3, 5, ... \quad (4.1)
\]
Figure 4.1. Left: Frequency analysis for a simple duct representing the downstream part of the long combustion chamber for different mixture temperatures $T_{mix}$ (without effects of reaction). Right: Eigenmode analysis solved for the extensive geometry with FFMLAB.

The frequencies of the azimuthal ($f_m$) (also called circumferential) and the radial ($f_n$) modes are derived in a similar way, whereby the characteristic length is defined by the radius of the duct $R$. $\beta_{mn}$ is the corresponding mode coefficient derived from the zero of the particular Bessel function (for further details see Section A.3.1. on page 167).

$$f_{m,n} = \frac{c}{2\pi R} \beta_{mn}$$  \hspace{1cm} (4.2)

Figure 4.1 shows the influence of an increasing air temperature (speed of sound) to the frequencies of the natural acoustic modes relevant for the further discussion. Higher modes, composed of several modes (e.g. $(1, 1, 0)$) are not shown to simplify matters.

In addition, a more adequate way to determine the resonant frequencies of a system has been used. The general description of a sound wave describing the spatial and temporal evolution of pressure fluctuations is given by the wave equation for acoustic waves and is straightforward derived from the equations of conservation (mass and momentum) ([9],[71],[75]). By considering only time harmonic solutions to solve for the wave equations of the form $p = p_0 \cdot \exp(i\omega t)$, a Helmholtz equation for acoustic waves can be stated. A source term (e.g. for the flame itself and/or for the representation of thermoacoustic driving sources) has been neglected in the current formulation (homogeneous Helmholtz equation). The Helmholtz equation is an adapted formulation to find eigenmodes and eigenvalues:

$$\nabla \cdot \left( \frac{1}{\rho_0} \nabla p \right) - \frac{\lambda p}{\rho_0 e^2} = 0$$  \hspace{1cm} (4.3)
where \( \rho_0 \) is the fluid density, \( p \) the pressure, \( c \) the speed of sound and \( \lambda \) represents the eigenvalue. The eigenfrequencies are calculated from the eigenvalues according to equation \( f = \sqrt[\lambda]{2\pi} \).

A finite element tool (FEMLAB), commercially available, has been used to implement the geometry of the gas turbine test bench according to figure 3.3 into a 3D FEMLAB model and to solve equation 4.3. Finite elements are automatically meshed with tetrahedral elements. The additional impedance due to the EV burner was not modelled nor the reaction zone with its inhomogeneous temperature distribution. The hard boundary condition has been used for all walls of the combustion chamber including the upstream section. The fluctuating pressure of the exit of the combustion chamber has been set to \( p = 0 \) (Boundary condition).

The lower eigenmodes for the combustion geometry implemented in FEMLAB and calculated at a temperature of the mixture \( T_{\text{mix}} = 700[K] \) are listed in Table 4.1. Major differences exist by comparing the higher azimuthal (>2) and radial modes of the eigenmode analysis with the resonant frequencies of the simple model (eq. 4.2), whereas the longitudinal modes almost coincide. Exemplary images of the eigenmode analysis are given in the appendix A.3.1.

<table>
<thead>
<tr>
<th>Eigenmode</th>
<th>Order</th>
<th>Eigenfrequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1\text{st} axial (longitudinal) mode</td>
<td>(1, 0, 0)</td>
<td>109[Hz]</td>
</tr>
<tr>
<td>2\text{nd} axial (longitudinal) mode</td>
<td>(2, 0, 0)</td>
<td>285[Hz]</td>
</tr>
<tr>
<td>3\text{rd} axial (longitudinal) mode</td>
<td>(3, 0, 0)</td>
<td>491[Hz]</td>
</tr>
<tr>
<td>4\text{th} axial (longitudinal) mode</td>
<td>(4, 0, 0)</td>
<td>650[Hz]</td>
</tr>
<tr>
<td>5\text{th} axial (longitudinal) mode</td>
<td>(5, 0, 0)</td>
<td>811[Hz]</td>
</tr>
<tr>
<td>1\text{st} azimuthal (transversal) mode downstream</td>
<td>(0, 1, 0)</td>
<td>3306[Hz]</td>
</tr>
<tr>
<td>1\text{st} azimuthal (transversal) mode upstream</td>
<td></td>
<td>3913[Hz]</td>
</tr>
<tr>
<td>2\text{nd} azimuthal (transversal) mode downstream</td>
<td>(0, 2, 0)</td>
<td>5472[Hz]</td>
</tr>
<tr>
<td>2\text{nd} azimuthal (transversal) mode upstream</td>
<td></td>
<td>6375[Hz]</td>
</tr>
<tr>
<td>1\text{st} radial mode downstream</td>
<td>(0, 0, 1)</td>
<td>7345[Hz]</td>
</tr>
<tr>
<td>1\text{st} radial mode upstream</td>
<td></td>
<td>8285[Hz]</td>
</tr>
</tbody>
</table>

Table 4.1. Eigenfrequencies for the first simple eigenmodes \((l, m, n)\) of the extensive geometry of the gas turbine combustor calculated at the mixture temperature \( T_{\text{mix}} = 700[K] \).
4.2. Flame structure

The current investigation covered a wide range of operating conditions over which the flame underwent visible and audible changes, such as flame shape, flame height, flame location and sound emissions.

For lean conditions (flame type $\text{I}_a$ and $\text{I}_b$ in Figure 4.2) combustion processes took place only in the central recirculation zone. The flame front was located 2 to 3 burner diameters downstream and the flame had a more or less open conical shape of 15 to 20 burner diameters height ("weakly burning" flame). For richer mixtures or higher mixture temperatures, the flame stabilized further upstream, close to the burner outlet, and extended into the outer recirculation zone (flame type $\text{II}_a$). The flame was approximately 1 to 2 burner diameters high and exhibited a compact envelope ("strongly burning" flame). The change of state from weakly to strongly burning flames exhibited two distinct steps as the mixture strength was increased: firstly, an abrupt change of flame shape and height attended by a significant increase of sound emission took place (the flame tip suddenly opened and extended into the outer recirculation zone, flame type $\text{II}_b$ to $\text{I}_b$). This "fast transition" happened for such small variations of air/fuel equivalence ratios as $\Delta \lambda_{afr} = 0.05[-]$, with an excellent repeatability and no hysteresis; secondly, the flame envelope was stretched in the streamwise direction and the flame seemed to jump back and forth between two stationary states characterized by a longer and a shorter flame height (flame type $\text{II}_a$). This transitional flame existed for a narrow band of parameters and is thus referred to as "slow transition" in the following. Finally, the flame became continuously more compact and reached the strongly burning state as $\lambda_{afr}$ was further decreased (flame type $\text{II}_a$).

The fast transition from flame type $\text{I}_b$ to $\text{II}_a$ has been previously observed experimentally ([31],[58],[78]). Giezendanner et al. [31] reported a similar transition in a 25[kW] counter-rotating swirl burner, while Polifke et al. [78] associated heat release instabilities with this transition caused by two different flame stabilizing mechanisms induced by non monotonic pressure drops across the burner. In this respect, the authors talk about a "bi-stable flame" due to pressure loss instability to do justice to its hydrodynamic nature. The second transition from flame type $\text{II}_a$ to $\text{II}_b$, or slow transition, has been observed on one hand experimentally by Broda et al. [11] and on the other hand numerically by Huang and Yang [44], who associated slow transition with flame speed perturbations enabling the flame to randomly change from one limit cycle, with small or no oscillations, to another limit cycle characterized by a different flame position and large oscillations, thus calling this transition a "flame bifurcation". In this context, turbulent flame speed variations, due to mixture temperature or equivalence ratio perturbations, allow the flame to flash back into the corner-recirculating zone and
drive flow oscillations through its influence on unsteady heat release. However, although the fast and slow transitions introduced in the previous paragraph have been observed individually with different burner configurations, the structure describing this two-steps transition from weakly to strongly burning flames has never been reported elsewhere. Furthermore, no detailed investigation on the stability properties associated with these transitions can be found in literature (appearance of new modes of instability, modifications of frequency and/or intensity for existing modes, etc.).

In fact, a fourth type of flame was observed for some particular operating conditions. For relatively low temperatures $T_{\text{mix}} \leq 600[K]$ and rich mixtures, the flame front suddenly moved further upstream into the burner attended by an audible deflagration. In this new regime, chemical reactions were mainly concentrated in a conical sheet in the burner cone and the sound produced by the flame was weaker for lower bands of frequencies. (flame (v)). However, since the flame front was lying on the burner walls, they became hot very fast and started to glue (rich limit). Thus, no systematic investigation was possible for this particular state with the current test facility.

The lean extinction limit and the fast transition occurred for significantly leaner mixtures as the temperature of the reactants were increased (see Figure 4.2). In counterpart, the extinction limit and the fast transition did not exhibit a significant dependence on the total mass flow rate and were only slightly shifted towards leaner mixtures with increasing $m_{\text{air}}$ (see Figure 4.3 where the flame structure is shown for $m_{\text{air}} = 30$ and $m_{\text{air}} = 46[g/s]$). The region of strongly burning flames (flame type (v)) extended towards leaner and colder mixtures for increasing $m_{\text{air}}$, pushing simultaneously the slow transition towards leaner mixtures and narrowing its region of occurrence. These observations tend to show that the extinction limit and the fast transition were rather independent from the burner exit velocities and mainly governed by chem-
Figure 4.3. Flame structure in the parameter space \((\lambda_{air}, T_{mix})\) for a constant mass flow rate 
\(m_{air} = 30\,[g/s]\) (left) and \(m_{air} = 46\,[g/s]\) (right) for the long combustion chamber (LCC).

Figure 4.4. Flame structure in the parameter space \((\lambda_{air}, T_{mix})\) for a constant mass flow rate 
\(m_{air} = 36\,[g/s]\) for the long combustion chamber (LCC, left) and the short combustion chamber (SCC, right).

The experiments leading to the complex flame structure with its two-steps transition from weakly to strongly burning flames previously described were repeated for the short combustion chamber (half length) and are given in Figure 4.4 for \(m_{air} = 36\,[g/s]\) (right). The lean extinction limit was slightly shifted towards leaner mixtures in comparison with the long combustion chamber and the fast transition stayed roughly unchanged. More strikingly, the slow transition was not observed for the short combustion chamber but a new, compact and noisy flame state was observed for hot and rich mixtures (flame type \(\circ\) in Figure 4.4). Although this new flame had a more compact flame envelope than flame type \(\bullet\) observed for the long combustion chamber,
its transitional nature was clearly visible and the flame was randomly jumping between two stationary states. Finally, the influence of the mass flow rate on the flame structure was also investigated for the short combustion chamber and the result showed an extension of the region corresponding to flame type \( \text{IIIB} \) towards leaner and colder mixtures, leaving lean extinction limits and the fast transition unchanged (See Figure 4.5).

**Figure 4.5.** Flame structure in the parameter space \( (\lambda_{afr}, T_{\text{mix}}) \) for a constant mass flow rate \( \dot{m}_{\text{air}} = 30 \text{[g/s]} \) (left) and \( \dot{m}_{\text{air}} = 46 \text{[g/s]} \) (right) for the short combustion chamber (SCC).

**Adiabatic flame temperature and Damköhler number compared to the transitions of the flame structure**

The flame structure depicted in Figure 4.2 for different mass flow rates revealed the dominant role of mixture temperature and composition on the occurrence of lean extinction limit and fast transition. A simple 0-dimensional model, including gas composition and a temperature-dependent specific heat \( c_p(T) \) of the mixture obtained with an iterative algorithm, was used to calculate the laminar adiabatic flame temperature \( T_{f, ad} \) as a function of \( \lambda_{afr} \) and \( T_{\text{mix}} \). The resulting iso-contours are shown in Figure 4.6 (left) for the long combustion chamber (LCC) and in Figure 4.7 (left) on page 58 for the short combustion chamber (SCC).

Iso-contours of \( T_{f, ad} \) coincided with the lean extinction limit and the fast transition, thus supporting the idea that both limits are mainly governed by thermochemistry rather than hydrodynamics, as already observed. In order to investigate the influence of flow velocity, the Damköhler number \( Da \) has been calculated as well. The Damköhler number is the ratio between a typical hydrodynamic time scale \( \tau_{\text{flow}} \) and a characteristic chemical time \( \tau_{\text{chem}} \). In the current investigation, \( \tau_{\text{flow}} = d_{EV}/u_0 \) where \( d_{EV} \) and \( u_0 \) were the burner diameter and the temperature-dependent velocity at the burner exit respectively, and \( \tau_{\text{chem}} \) was obtained from CHEMKIN. The resulting iso-contours of
the Damköhler number are shown in Figure 4.6 and 4.7 (right). Damköhler iso-contours exhibit steeper slopes than adiabatic flame temperatures, illustrating the effect of $T_{\text{mix}}$ on the exit velocity $u_0$ (the total mass flow rate was kept constant). The most unstable flames (Lb) and (Lb) for the long and short combustion chamber respectively exhibit similar slopes that might illustrate the relevance of both chemical and hydrodynamic parameters on these flames. However, hydrodynamic quantities like velocity profiles at the burner exit and recirculation zone sizes, among others, should be known to further confirm this trend and identify key parameters governing the flame structure.
4.3. Pressure drop

The pressure drop across the flame measured for all flame states observed in the current investigation exhibited significant differences. It have to be assumed, that the size and strength of the swirl-induced recirculation zones, controlling the effective burner exit area, were strongly influenced by the flame shape and position. In turn, recirculation zones are expected to be affected by the density change across the flame which decreased the local swirl number and consequently modified radial pressure distribution. This phenomenon is analyzed in more details in this section.

Averaged static pressures were measured up- and downstream of the burner for all operating conditions. In order to extract the pressure drops $\Delta P_{\text{flame}}$ caused only by the flame, these measurements were performed for non reacting flows as well (all other parameters were kept identical). Assuming that flow properties were not basically altered due to a reaction zone, the pressure rise at the burner produced by the flow only depends on the flow velocity and the density of the flow (expressed here in terms of mixture temperature $T_{\text{mix}}$ and mass flow rate $m_{\text{air}}$). This back pressure $\Delta P_{\text{flow}}$ is registered on the EV burner only by non-reactive flows through the burner and is depicted on the right side of Figure 4.8. The measured pressure drop $\Delta P_{\text{tot}}$ for the reactive case includes the pressure drops of the flame $\Delta P_{\text{flame}}$ and the back pressure of flow through the burner $\Delta P_{\text{flow}}$. In order to obtain pressure drops engendered by the flame, the back pressure of the combustor is subtracted from static pressure measurements of the reacting experiment ($\Delta P_{\text{flame}} = \Delta P_{\text{tot}} - \Delta P_{\text{flow}}$). The result is shown in Figure 4.10 for both combustion chamber lengths (LCC and SCC) and for a total mass flow rate of $m_{\text{air}} = 36$ [g/s].

![Figure 4.8](image-url) Left: Iso-contours of the pressure drop across the flame including the back pressure resulting from the flow through the EV burner in the $(\lambda_{\text{flame}}, T_{\text{mix}})$ plane for $m_{\text{air}} = 36$ [g/s] and for the long combustion chamber (LCC). Right: Back pressure $\Delta P_{\text{flow}}$ of the combustor measured for the non reacting flow in the range $m_{\text{air}} = [30 - 46]$ [g/s].
Pressure drop

Figure 4.9. Left: Iso-contours of the pressure drop across the flame in the \((\lambda_{af}, T_{mix})\) plane for \(m_{air} = 36[g/s]\) and for the long combustion chamber (LCC). Right: Pressure drop across the flame as a function of \(\lambda_{af}\) for different \(T_{mix}\) and for the same conditions as left.

The connection between the flame structure discussed in the previous section and the pressure drops across the flame is straightforward. For both combustion chamber lengths, pressure drops exhibited a significant increase for transitional flames (flame type \(\bullet\) and \(\bigcirc\) for the long and short combustion chamber, respectively). Polifke et al. \[78\] reported similar observations and derived a physical mechanism for heat release instabilities associated with negative slopes of pressure drop - mass flow rate characteristics.

A similar mechanism can be derived from the current measurements focusing on the negative slopes of the pressure drop - air/fuel equivalence ratio characteristics (see Figure 4.9, right). Assuming a negative perturbation of the air/fuel equivalence ratio,

Figure 4.10. Iso-contours of the pressure drop across the flame in the \((\lambda_{af}, T_{mix})\) plane for \(m_{air} = 36[g/s]\). Left: Long combustion chamber (LCC). Right: Short combustion chamber (SCC).
the flame front will move upstream towards the burner. For flames where \((d(Δp)/dλ) ≤ 0\) holds (flame (1b) and (1b)), the pressure drop caused by the flame will simultaneously increase, decreasing the mass flow rate and thus enabling the flame front to move even further against the flow. However, once the flame passed the streamwise coordinate corresponding to the maximum of pressure drop, the total mass flow rate will increase again and the flame front will move back to its initial location corresponding to its average air/fuel equivalence ratio. The region of axially stretched flames (1b) and (1c) coincided with the maximum of pressure drop and pressure fluctuations (shown in the next section) and thus tend to support this mechanism. Furthermore, this flame behaviour is attended by large heat release variations able to induce thermoacoustic instabilities when the acoustic feedback satisfies the correct phase relation.

4.4. Dynamic pressure

In order to identify unstable flames and to quantify the amplitude of pressure oscillations in the combustion chamber, the normalized root mean square of the dynamic pressure \(p_{rm}\) was calculated for all flame types (See appendix A.2.2). Figure 4.11 and 4.12 (left) show the iso-contours of the SPL \(p_{rm}\) in [dB] as a function of mixture temperatures \(T_{mix}\) and air/fuel equivalence ratios \(λ_{afr}\) for a constant mass flow rate of \(m_{air} = 36 [g/s]\) and for both combustion chamber lengths. The right side of Figure 4.11 and 4.12 shows \(p_{rm}\) as a function of \(λ_{afr}\) for different (constant) mixture temperatures.

The amplitude of pressure oscillations rose significantly as the fast transition took place and reached a maximum for slightly richer mixtures. As expected from previous obser-
Figure 4.12. Left: Iso-contours of $p_{rms}$ in the ($\lambda_{sfr}, T_{mix}$) plane measured with microphone no.2 (MIC2) for $\dot{m}_{sfr} = 36[g/s]$ in the short combustion chamber (SCC).
Right: $p_{rms}$ as a function of $\lambda_{sfr}$ for different mixture temperatures $T_{mix}$.

...vations, the maximum of acoustic energy measured in the combustion chamber coincided with the maximum pressure drop (see Figure 4.10 on page 60) and corresponded to flames in the slow transition region described earlier. For lower mixture temperatures $T_{mix}$ in the range of 500 - 650[K], the flame type (ıa) close to the lean extinction limit is clearly distinguishable from flame type (ıb). However, the apparently large difference of 20[dB] in the SPL levels corresponds only to an absolute pressure difference of 12[Pa] between the two flame structures.

The measurements of $p_{rms}$ obtained with the short combustion chamber exhibited a similar behaviour (see Figure 4.12), with the maximum of pressure oscillations coinciding with flames in the slow transition. However, the increase of $p_{rms}$ with $T_{mix}$ and $\lambda_{sfr}$ of the short combustion chamber exhibited two distinct slopes with a plateau in between. The two slopes corresponded to the fast and slow transitions which are clearly distinct for the short combustion chamber, unlike in the long combustion chamber where the plateau is not visible.

4.4.1. Photomultipliers signal vs. dynamic pressure signal

Measurements on pressure fluctuations suggest that the most unstable flames were connected with a critical flame position, with respect to the recirculation zones, rather than with the distance between the flame and burner. Indeed, the normalized mean chemiluminescence intensity of the OH radical, measured on the axis of symmetry one diameter downstream of the burner, exhibited a rapid increase after the fast transition (See Figure 4.13 on page 63, left) followed by a plateau where the OH-chemiluminescence intensity oscillated around a constant value (See Figure 4.13, right for the root mean
Figure 4.13. Left: Iso-contours of the normalized averaged OH-chemiluminescence measured with the photomultiplier one burner diameter downstream of the dump plane on the axis of symmetry. Right: Iso-contours of the centered and normalized root mean square OH-chemiluminescence signal (same conditions as left, both at $\dot{m}_{air} = 36[g/s]$).

square values) and finally the intensity monotonically increased up to the rich limit. This plateau is an indication of the transitional region between two different limit cycles with distinct characteristics (flame structure $\square$ and $\square$).

The pressure drop across the flame, the dynamic pressure and the OH-chemiluminescence intensity have been discussed individually in function of $\lambda_{afr}$ and $T_{mix}$. In order to summarize these results and illustrate the connection between these quantities, $\Delta p_{flame}$, sound pressure level SPL ($p_{rms}$), $PM_{mean}$ and $PM_{rms}$ (mean and root mean square of the photomultiplier signal respectively) are shown in Figure 4.14 on page 64 as a function of $\lambda_{afr}$ for $T_{mix} = 650[K]$ and $\dot{m}_{air} = 36[g/s]$.

The fast (flame type $\square$ to $\square$) and slow (flame $\square$) transitions are clearly visible in OH chemiluminescence ($PM_{mean}$) as mentioned. The root mean square of the OH-chemiluminescence intensity exhibited two distinct steps corresponding to both transitions as well. The Sound Pressure Level (SPL) was hardly affected by the fast transition and increased continuously from lean flames towards richer flames to reach its maximum for flame type $\square$. The pressure drop caused by the flame was not affected at all by the fast transition and increased significantly only during the second slow transition.

These observations suggest that the second slow transition played a decisive role for the amplitude of pressure oscillations compared to the abrupt flame transition. The maximum of $PM_{rms}$ is clearly connected to the rapid increase of $\Delta p_{flame}$, but, unlike the observation of Polifke et al. [78], it is associated with the transitional flame $\square$ and is hardly affected by the fast flame transition.
4.4.2. Frequency content of dynamic pressure

In order to analyse the frequency content of pressure fluctuations in the combustion chamber, a fast Fourier transform was performed on the acquired pressure signals. Figure 4.15 shows typical spectra obtained for similar flame types, i.e. flame type (a) for the short (upper curve) and (b) for the long (lower curve) combustion chamber, as a function of the non-dimensional frequency in terms of Strouhal numbers $St$ (the Strouhal number is defined as $St = f \cdot \frac{d_{EV}}{u_0}$, where $f$ is the dimensional frequency, $d_{EV}$ the burner diameter and $u_0$ the average velocity at the burner exit). The dimensional frequency $f$ is indicated on top of each figure for information. The left side of Figure 4.15 shows the entire spectrum while the right side shows the low frequency range $St = [0 - 2]$ in more details.

Fundamental acoustic modes at $St = 0.30$ and $St = 0.65$ for the long and short combustion chamber respectively, and their higher orders at $St = 0.90$ and $St = 1.50$ for the long as well as $St = 1.95$ and $St = 3.25$ for the short combustion chamber are clearly visible in Figure 4.15. Their frequencies correspond to the first (quarter wave), second and third longitudinal acoustic modes ($\odot$) of the combustion chamber. The high frequency bands around $St = 5.80$ and $St = 9.00$ can be associated with the first and second azimuthal modes ($\odot$) derived with the finite element method in the downstream section and are equal for both combustion chamber lengths (within the small temperature difference). Moreover, the first harmonic ($\odot$) of the fundamentals at $St = 0.65$ and $St = 1.30$ for the long and short combustion chamber respectively are
Figure 4.15. Pressure spectrum as a function of Strouhal numbers $St$ for $\lambda_{afr} = 2.0$, $T_{mix} = 700[K]$ and $\dot{m}_{air} = 36[g/s]$, showing flame type (a) in the long (LCC) and the short combustion chamber (SCC). Left: $St = [0-10]$, right: $St = [0-2]$.

clearly visible as well. Note that these modes are all present in the system because the acoustic feedback was the most efficient at these discrete frequencies. However the onset of thermoacoustic instabilities in gaseous media usually involves other underlying mechanism coupling acoustic and heat release oscillations.

In order to obtain an in-depth analysis of the influence of $\lambda_{afr}$ (and of the flame types as well) on the excited modes, the iso-contours of the pressure amplitude were plotted as a function of $\lambda_{afr}$ and Strouhal number $St$ in Figure 4.16 for a constant mass flow rate of $\dot{m}_{air} = 36[g/s]$, a temperature of 700[K] and for the long combustion chamber. Figure 4.16 shows the frequency range up to $St = 10$ on the left side and Figure 4.16 right shows the low frequency range $St = [0–2]$ in more details.

Figure 4.16. Iso-contours of the pressure amplitudes (spectra) as a function of $\lambda_{afr}$ and Strouhal number $St$ for the long combustion chamber (LCC) operated at $T_{mix} = 700[K]$ and $\dot{m}_{air} = 36[g/s]$. (Left: $St = [0–10]$, right: $St = [0–2]$).
Figure 4.16 offers the possibility to track unstable modes and check their existence over the entire range of investigated equivalence ratios. The observations made out of Figure 4.16 are the following. Firstly, only 2 high frequency modes with Strouhal numbers equal to 1.5 and between 4 and 6.5 had a significant amplitude for all flame types, in particular for weakly burning flames (the fast transition occurred at $\lambda_{afr} = 2.25[-]$ for the same conditions, see Figure 4.18). Secondly, the mode with

![Pressure spectrum as a function of the Strouhal number $St$ for $\dot{m}_{CH4} = 361g/s$, $T_{air} = 650[K]$ and for typical $\lambda_{afr}$ values corresponding to the flame types investigated in the long combustion chamber (LCC) (Left: $St = [0 - 10]$, right: $St = [10 - 20]$).](image)
$St = 1.50$ had a more or less constant normalized frequency ($1.4 \leq St \leq 1.6$) for all mixtures, while the mode with $4 \leq St \leq 6$ (associated with the $1^{st}$ azimuthal acoustic mode) exhibited a continuous drift and an abrupt change coinciding with the fast transition previously described. This constant drift is an indication of the mode’s temperature dependence and the speed of sound rather than fluid velocity should be used for normalization for this mode. On the contrary, the constant Strouhal numbers of low frequency modes seemed to point to an underlying mechanism independent from flame and combustion chamber temperatures. Thirdly, low frequency modes $(St \leq 1.4)$ appeared and grew very rapidly once the fast transition took place (coming from lean to rich flames). These richer flames were much shorter than before the fast transition occurred and thus could more easily transfer energy into the acoustic modes of the combustion chamber (increased Rayleigh index). Finally, the dominant mode at $St = 0.3$ reached its maximum at $\lambda_{sfr} = 2.15$ corresponding to the transitional flame type (Figure 4.16). After having passed this maximum, the amplitude of the dominant mode decreased towards richer mixtures, and, simultaneously, modes with higher frequencies (at $St = 0.45, 0.9, 1.5$ and between $4$ and $6.5$) rapidly grew for flame type IIIa. Also, note that right at the fast transition, lower frequency bands $(0.05 \leq St \leq 0.3)$ were excited as well but the acoustic energy was distributed over a continuum rather than at discrete frequencies.

Figure 4.17 shows several pressure spectra for different air/fuel equivalent ratios $\lambda_{sfr}$ each corresponding to typical flame structure and in Figure 4.18, the flame types are given accordingly to identify the flame structures present in the long combustion chamber ($T_{mix} = 650[K]$).

![Figure 4.18](image_url)

**Figure 4.18.** Iso-contours of the pressure amplitudes (spectra) as a function of $\lambda_{sfr}$ and Strouhal number $St$ for the long combustion chamber (LCC) operated at $T_{mix} = 650[K]$ and $m_{fuel} = 36[g/s]$ and corresponding numbers of the flame types to identify the flame structures (Left: $St = [0 - 10]$, right: $St = [10 - 21]$).
The influence of the total mass flow rate on the unstable modes was investigated in the same manner and is shown in Figure 4.19 where the iso-contours of the pressure amplitude are plotted for mass flow rates of \(30 \text{[g/s]} \) (left) and \(46 \text{[g/s]} \) (right) and a temperature of \(700\text{[K]} \).

The comparison between Strouhal numbers of excited modes for both mass flow rates indicate that their dimensional frequency has not been increased proportionally to the mass flow rate, except for the mode at \(St = 1.5\), which Strouhal number stayed constant. This mode scaled with the mean axial velocity at the burner outlet supporting the idea of a hydrodynamic mode. All other excited frequency bands were associated with acoustic eigenmodes of the combustion chamber for which the feedback of acoustic energy was at its maximum.

However, all acoustic modes did not exhibit the same dependence on temperature. The higher frequency modes \((St \geq 3)\) associated with the circumferential length were strongly affected by temperature while this effect on low frequency modes (longitudinal modes) was negligible. The larger mass flow rate globally increased sound pressure levels without changing the frequency content. Finally, the lean extinction limit has been shifted towards leaner mixtures with increasing mass flow rates which is probably due to modifications of the recirculation bubble size and the higher power density at higher mass flow rates, making the flame less susceptible for disturbances resulting in flame blow off.

The frequency content of pressure measured with the short combustion chamber was analyzed as well and the corresponding picture is shown in Figure 4.20 for a constant mass flow rate of \(36\text{[g/s]}\) and a temperature of \(700\text{[K]}\). The high frequency range
Figure 4.20. Iso-contours of the pressure amplitudes (spectra) as a function of $\lambda_{afl}$ and Strouhal number $St$ for the short combustion chamber (SCC) operated at $T_{\text{mix}} = 700K$ and $\dot{m}_{afl} = 36g/s$. (Left: $St = [0-10]$ , right: $St = [0-2]$).

($St \geq 1.5$), associated with azimuthal acoustic modes, stayed unchanged as expected while the low frequency range has been significantly modified by combustion chamber length and reflected the flame structure depicted in Figure 4.3 on page 56. Indeed, the fast transition at $\lambda_{afl} = 2.3$ is clearly visible and the modes at $St = 0.3$, $St = 0.6$ and $St = 2.0$ grew rapidly once it occurred. These modes reached their maximum for the transitional stretched flame (\text{\textit{b}}), as for the long combustion chamber. However, this flame type was reached for richer mixtures (its onset is clearly visible at $\lambda_{afl} \leq 1.9$ in Figure 4.20) and the resulting flame envelope was thus slightly compacter at the onset of thermoacoustic instabilities for the short combustion chamber, in comparison with the long one, as expected from Rayleigh criteria.

The iso-contours of the pressure spectra for mass flow rates ranging from 30 up to $46g/s$ are listed in detail in the appendix “Iso-contours of the SPL for constant mass flow rates” on page 171 for the different investigated locations of the microphones (MIC1-MIC3).
4.4.3. Dominant mode analysis

The pressure spectra showed that the acoustic energy was mainly concentrated in a few distinct frequency bands rather than distributed over a broadband spectrum. These modes were analyzed individually as a function of air/fuel equivalence ratios for fixed mixture temperatures and mass flow rates. The individual spectra (for a given $\lambda_{afr}$) were first filtered with a Savitzky-Golay filter and the noise floor was calculated with a SNIP algorithm and subtracted from the filtered spectra (See Section 3.5.2.2). Dominant modes were then identified, tracked along its maxima in the iso-contour representation and integrated over the peak width. Finally, the contributions from each mode were normalized with the sum, resulting in a contribution per mode, in percent, for each operating point. Instability modes used for this calculation are shown in Figure 4.21 (left) as a function of air/fuel equivalence ratio and Strouhal numbers and the normalized contribution of each mode is shown in the same figure (right).

The dominance of the mode associated with the 1st longitudinal acoustic mode in the range of $1.9 \leq \lambda_{afr} \leq 2.6$ is clearly visible in Figure 4.21. The maximum of this mode corresponds to the transitional flame (III). For lean mixtures above $\lambda_{afr} = 2.6$ (the flame is lifted and elongated with a “V” shape), the relative amplitude of the 3rd longitudinal mode slightly grew and the 2nd azimuthal mode became dominant. For very low air/fuel equivalence ratios ($\lambda_{afr} \leq 2.1$), the amplitude of the 1st longitudinal mode decreased rapidly while all higher orders globally won weight.

The same analysis was done with the short combustion chamber and is shown in Figure 4.22. Unlike with the long combustion chamber, two modes at $St = 0.3$ and

![Figure 4.21](image_url)
$St = 0.6$ were clearly dominant. The relative amplitude of the lower frequency mode at $St = 0.3$ was roughly doubled for leaner condition ($\lambda_{afr} \geq 2.2$) and the mode at $St = 0.6$ corresponding to the 1st longitudinal acoustic mode of the combustion chamber grew significantly for richer air/fuel equivalence ratios. This mode became clearly dominant for $\lambda_{afr} \leq 2.0$. For lean conditions, the high frequency mode associated with the 1st azimuthal mode became dominant like for the long combustion chamber.

A similar investigation was made by Poinsot et al. [77] in a multiple inlet dump combustor. The authors plotted constant iso-contours of the acoustic spectral density of each unstable frequency band as a function of $\lambda_{afr}$ and $\dot{m}_{air}$ thus identifying the range of occurrence of each oscillation. This method did not account for a behaviour, where modes are changing its frequencies more than the previously selected bandwidth (e.g. by tracking the maxima of a mode).

However, their measurements showed that the domains corresponding to each mode were mutually exclusive with a general trend towards higher acoustic intensities and higher frequencies with increasing mass flow rate and mixture strength. In the current investigation, high frequency modes had also a tendency to increase with growing mixture's richness, but to a smaller extent. Indeed, the acoustic intensity globally increased for all modes with increasing richness so that the normalized contribution of the high frequency ones stayed small with respect to low frequency modes. Moreover, unlike the investigation of Poinsot et al., several unstable modes could exist simultaneously in the current investigation, in particular for the short combustion chamber.

![Figure 4.22](image_url)

**Figure 4.22.** Left: Dominant instability modes shown in the plane ($\lambda_{afr}$, $St$) for $T_{mis} = 700[K]$ and $\dot{m}_{air} = 36[g/s]$ (SCC). Right: Normalized contribution of each individual mode as a function of $\lambda_{afr}$ (all other conditions are the same as left). Flame types with their symbols are given for information.
4.5. Non-reacting and reacting flow: A comparison

The influence of hydrodynamic stability on thermoacoustic properties of the flame has been pointed out by many authors ([31],[74],[77],[94]). In this section, acoustic characteristics of the non-reacting flow are analyzed in more details for the same mass flow rates, mixtures and temperatures as for the reacting flows. The result obtained for an air/fuel equivalence ratio of 2.0 and a mass flow rate of 36[g/s] is shown in Figure 4.23 left as a function of mixture temperature and Strouhal number (the reacting case is shown right for comparison). Note that the total mass flow rate was kept constant and the velocity used for the Strouhal number calculation was corrected for temperature using the ideal gas law.

Three different frequency bands were excited for all mixtures temperatures (exit velocities). The modes at $St = 0.3$ and $St = 1.5$ were present for all operating conditions with a constant normalized frequency for all investigated temperatures and combustion chamber lengths thus scaling with the mean velocity at the burner exit. Their frequency was a function of initial velocity but was not affected by the temperature in the combustion chamber or its length for example. These observations tend to support the existence of an underlying hydrodynamic mechanism inducing and triggering the instability for these two modes.

Paschereit et al. [74] measured velocity profiles for a similar large-scale EV burner in a water channel and found two unstable modes at $St = 0.58$ (axisymmetric) and $St = 7.77$ (helical) associated with the outer shear layer (jet-like profile) and one unstable mode at $St = 1.2$ (helical) associated with the inner shear layer (wake-like profile).

Figure 4.23. Iso-contours of the pressure amplitudes (spectra) as a function of $T_{mix}$ and Strouhal number $St$ for the long combustion chamber (LCC) operated at $\dot{m}_{fuel} = 36$[g/s]. Left: Non-reacting flow $\lambda_{fuel} = \infty$. Right: Reacting flow at $\lambda_{fuel} = 2.0$. 

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The frequency of shear-driven instabilities is usually very sensitive to changes of the velocity gradient in the mean profile and these results can not be compared one to one to the Strouhal numbers found in the current investigation. However, their similarities provide an additional indication of its hydrodynamic nature. Finally, the high frequency mode \( St = [4 - 6] \) was associated with the 2\(^{nd} \) azimuthal acoustic mode and scaled rather with the speed of sound.

4.6. Emissions

The emission performance of the EV5 burner is in a similar order of magnitude compared to an atmospheric fired EV17 combustor [48]. Pollutants can be characterized by a negligible soot formation tendency, a low level of \( CO \) and a close to zero level of all unburnt hydrocarbons \( UHC \) as well as a \( NO_x \)-production rate in the single digit range. In addition, carbon monoxide \( CO_2 \), water vapour \( H_2O \) and excess oxygen \( O_2 \) are present in the exhaust. Oxides of sulfur \( SO_x \) play a secondary role, since its formation is a strict consequence of the appearance of sulfur in the fuel.

To compare experimental data of combustors running at various air/fuel equivalence ratios and to remove the influence of humidity, emission values are rescaled (converted from a wet to a dry basis) and referenced to 15\% oxygen measured on a dry basis. The equation to correct the numerical values of an actual measured emission concentration \( c_{emiss} \) (in [ppm]) of a dry basis is given by

\[
\frac{c_{emiss,\, dry,\, ref\, 15\%\, oxygen}}{c_{emiss,\, dry}} = \frac{(20.9 - 15) \cdot c_{emiss,\, dry}}{20.9 - c_{O2,\, dry}},
\]

whereby the actual concentration of oxygen \( c_{O2,\, dry} \) is present in the exhaust and expressed in percentage by volume. An important aspect of the 15\%-\( O_2 \) reference is the dependency of the emission concentration on the accuracy of the actual oxygen measurement. An improper, measured value of oxygen results in a misleading concentration of the pollutant.

4.6.1. Emission measurements

The main focus of conducting emission measurements within the current investigation is to correlate pressure fluctuations and flame structures with emission information. In order to summarize these results and illustrate a connection between these quantities, emissions, whereby \( NO_x \), \( CO \) and \( UHC \) are expressed in [ppmv] and \( CO_2 \) as well as \( O_2 \) are stated in [Vol.-\%] are shown in Figure 4.24 on a dry basis as a function of \( \lambda_{af} \)
Emissions

for $T_{mix} = 650[K]$ and $m_{air} = 36[g/s]$. In the background of the figure, corresponding ranges of the air/fuel equivalence ratio for the different flame structures are given as well. The lower part of Figure 4.24 shows a similar representation as the upper part unless the fact that the values for $NO_x$, $CO$ and $UHC$ are referred to $15\%$ oxygen. For operating conditions leaner than 2.4 ($\lambda_{afr} > 2.4$), the varying flame structure tends to result in an inaccurate emission measurement. This weak flame enabled the flow to establish a recirculation zone at the atmospheric access at the end of the combustion chamber. Oxygen of the ambient air was transported into the downstream part of the combustor and the emission probe located close to its top was measuring diluted exhaust gases for these operating conditions and above. Extensive tests to place the probe at various locations in the combustor as well as modifying the probe itself have been made to exclude this effect. Other positions showed other drawbacks. Major

![Figure 4.24](image-url)

Figure 4.24. Upper figure: The left axis correspond to the measured $NO_x$, $CO$ and $UHC$ emissions, where the right axis belongs to $CO_2$ and $O_2$ values in the long combustion chamber (values are a function of $\lambda_{afr}$ for $T_{mix} = 650[K]$ and $m_{air} = 36[g/s]$). Lower figure: The conditions are the same as above and the $NO_x$, $CO$ and $UHC$ emissions are scaled to the reference of $15\%$ $O_2$.
importance was given to achieve representative measurements at operating conditions of $\lambda_{afr} < 2.4$ rather than to investigate leaner conditions. Therefore, the position and the sensing element itself were optimised to the band of equivalence ratios of interest ($\lambda_{afr} = [1.7 - 2.4]$).

In Figure 4.24 the $NO_x$ emissions increase inverse proportional with the air/fuel equivalence ratio since the flame temperature of the primary reaction zone is a function of $\lambda_{afr}$ and $T_{mix}$ (The additional influence of the mixture temperature $T_{mix}$ to the $NO_x$ production rate is shown in Figure 4.25). A sudden step at $\lambda_{afr} = 2.45$ of the $NO_x$ values (at a 15%-O$_2$ reference) to a higher level is a scaling artefact of the diluted O$_2$ measurement. The abrupt change in the curve progression of CO, UHC and CO$_2$ is traced back to the similar fact. The fully premixed combustor and the excess of oxygen yield too small values of CO and UHC. The slight increase of CO and UHC emissions for lean conditions ($\lambda_{afr} = [1.7 - 2.3]$) derive from a cooler reaction zone at leaner equivalence ratios. Above air/fuel equivalence ratios of 2.2, an adequate burning rate is incrementally degraded and results in a steeper rise of CO and UHC emissions. Emission maps, similar to these $NO_x$-maps depicted in Figure 4.25 are additionally given in the appendix “Emissions” on page 169 for CO, UHC, CO$_2$ and O$_2$.

Several observations were made regarding the appearance of the flame structure and the corresponding emissions. First of all, there is no visible tendency of combustion oscillation to increase the $NO_x$ levels or to cause an incomplete combustion and to produce higher levels of unburnt hydrocarbons or carbon monoxide. The slow transition of the flame structure seems to be of no significance for any investigated emission nor the pressure fluctuations have any impact (The red lines in Figure 4.24 indicate the air/fuel equivalence ratio where the maximum value of the sound level pressure was measured.

![Figure 4.25](image-url)

**Figure 4.25.** Left figures: CO, UHC and NO$_x$ emissions shown at a dry basis in the ($\lambda_{afr}, T_{mix}$) plane for a constant mass flow rate $m_{air} = 36\,[g/s]$. Right figures: Emissions are shown at a reference of 15%-O$_2$ (same conditions as left).
Emissions within the \( \lambda_{\text{afr}} \) range. A steeper slope of \( NO_x \) levels is observed along with the strongly burning flames of flame type (iiia) (See Figure 4.25). These flames are characterized by a compacter flame shape located closer at the burner exit. Thus, the reaction zone of this type of flame has locally a higher temperature, by what the formation of nitrogen oxide is encouraged.

In contrast to the slow transition, the fast transition from flame type (ib) to (iiia) is characterized by a CO emission drop. The reaction zone of (ib) seems to be too cold to oxidize completely carbon monoxide to carbon dioxide even though an excess of oxygen is present (The adiabatic flame temperature \( T_{f,ad} \) is lower than 1650[K] at the transition). By further reducing the chemical energy fed into the system (going to leaner conditions), the combustion efficiency is considerably reduced by a chilling effect and UHC emissions rise to an unwanted high level (Figure 4.24). This chilling effect is related to the region where flame type (ib) has already been defined.

4.6.2. Trade-off: Flame stability vs. nitrogen oxides

The parameters defining an operating condition and their influence on the formation of \( NO_x \) and the susceptibility to thermoacoustic instabilities expressed in terms of the sound pressure level measured with microphone no.2 is summarized in Figure 4.26. Starting at very low \( NO_x \) and sound pressure levels for the flame types (ib) and (ia), a medium decrease of the air/fuel equivalence ratio \( \lambda_{\text{afr}} \) will change the flame structure after passing the fast transition line to flame type (iiia). Although the chemical power density of the flame is increased, the emission level as well as the sound pressure level persist at a reasonable low magnitude.

**Figure 4.26.** \( NO_x \)-levels and sound pressure levels of microphone no. 2 showing the trade-off stability vs. emission for diverse operating conditions all within a constant mass flow rate \( m_{\text{air}} = 36[g/s] \) and whereby connected curves correspond to constant mixture temperature \( T_{\text{mix}} \).
By an additional decrease of $\lambda_{afr}$ to richer conditions, the sound pressure level rises dramatically and the slow transition of flame type 12 is achieved. The mixture temperature mainly affects the sound pressure level at which the inflexion point of the slow transition is reached and according to Figure 4.11, it can be stated that, by varying the air/fuel equivalence ratio, higher mixture temperatures feature lower sound pressure levels within the slow transition than colder mixtures. Therefore, a colder mixture temperature causes an inflexion point at higher sound pressure levels (curves are shifted). By an auxiliary drop of $\lambda_{afr}$, pressure oscillations are lowered and the formation of $NO_x$ remains almost similar. However, for very rich operating conditions, the $NO_x$-level starts steeply rising and the sound pressure is levelling off at a value of $147[dB]$ with an asymptotic behaviour. An increasing mass flow rate $m_{air}$ causes a shift to higher $NO_x$-levels and to less enlarged pressure oscillations.
Seite Leer / Blank leaf
This chapter focuses first on time-averaged OH-chemiluminescence and OH-PLIF images for the flame structure, followed by a detailed analysis of the effects of air-to-fuel ratio, mixture temperature and mass flow rate on the mean flame front and location of the flame gravity center. Phase-averaged images of the flame are shown for unstable operating conditions and the corresponding evolution of flame front, center of gravity (CoG) and intensity are analyzed for an instability period.

5.1. Flame structure

The current parametric investigation covered a wide range of operating conditions over which the flame underwent visible and audible changes, such as flame shape, flame height, flame location and sound emissions. Different flame types have been described and characterized basically in chapter 4. However, an optical investigation of the individual flame structures is still missing. To this end, the OH chemiluminescence and OH-PLIF images were taken for representative operating conditions at a constant mass flow rate $\dot{m}_{air} = 40 \text{[g/s]}$.

![Figure 5.1. Flame structure and investigated flames in the parameter space $(\lambda_{air}, T_{mix})$ and for a constant mass flow rate $\dot{m}_{air} = 40 \text{[g/s]}$.](image)  
Left: Long combustion chamber (LCC). Right: Short combustion chamber (SCC).  
$\times$: time-averaged (not synchronized) measurements  
$\bigcirc$: time-averaged + phase-averaged (phase-locked) images  
$: time-averaged + phase-averaged images for $\dot{m}_{air} = [30, 36, 40, 42, 48] \text{[g/s]}$
flow rate of $\dot{m}_{air} = 40 \text{[g/s]}$ within the parameter space $(\lambda_{air}, T_{mix})$ and indicated in Figure 5.1 with crosses $\times$, circles $\circ$ and squares $\square$. Crosses $\times$ indicate time-averaged (not synchronized) measurements, whereby for circles $\circ$ phase-averaged (phase-locked) images were additionally taken. Moreover, $-$symbols point out the supplementary time- and phase-averaged images recorded for the additional mass flow rates of $\dot{m}_{air} = [30, 36, 40, 42, 48] \text{[g/s]}$.

### 5.1.1. Characterization of the different flame types

Flame shapes of all investigated flame types (la) - (lb) are depicted within this subsection. The intensity images of the OH chemiluminescence are exemplarily shown for the different flame types appearing in the long combustion chamber in Figures 5.2 and 5.4. For the images on the left side of the figures, the intensity is normalized with the max-

![Figure 5.2. Time-averaged images of the OH chemiluminescence intensity investigated in the long combustion chamber (LCC).](image)

Left: Normalization of the intensity with the individual maximum of each image. Right: Normalization of the intensity with the absolute maximum of all images.

Top: Flame type (la) at $\lambda_{air} = 2.8$, $T_{mix} = 700 \text{[K]}$, $\dot{m}_{air} = 40 \text{[g/s]}$.

Bottom: Flame type (lb) at $\lambda_{air} = 2.6$, $T_{mix} = 700 \text{[K]}$, $\dot{m}_{air} = 40 \text{[g/s]}$. 

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Flame structure

maximum of each individual image, whereas for images shown on the right side, the normalization is performed with the maximum intensity value obtained within all OH chemiluminescence images acquired. The intensity images of the OH-PLIF are similarly presented in Figures 5.3 and 5.5 for the same operating conditions and the images are normalized respectively.

For lean conditions (flame type (ia) and (ib)) the combustion processes only take place in the central recirculation zone and the flame has a more or less open conical shape of 15 to 20 burner diameters height (“weakly burning” flame). For richer or hotter mixtures, the flame stabilizes closer to the burner outlet and extends into the outer recirculation zone (flame type (ili)). The flame is approximately 1 to 2 burner diameters high and exhibits a compact envelope (“strongly burning” flame). The transition from weakly to strongly burning flames takes place in two distinct steps: firstly, an abrupt

![Figure 5.3. Time-averaged images of the OH-PLIF investigated in the long combustion chamber (LCC). Left: Normalization of the intensity with the individual maximum of each image. Right: Normalization of the intensity with the absolute maximum of all images. Top: Flame type (ia) at $\lambda_{eff} = 2.8$, $T_{mix} = 700\, [K]$, $\dot{\omega}_{dir} = 40\, [g/\ell]$. Bottom: Flame type (ib) at $\lambda_{eff} = 2.8$, $T_{mix} = 700\, [K]$, $\dot{\omega}_{dir} = 40\, [g/\ell]$.

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Figure 5.4. Time-averaged images of the OH chemiluminescence intensity investigated in the long combustion chamber (LCC).
Left: Normalization of the intensity with the individual maximum of each image.
Right: Normalization of the intensity with the absolute maximum of all images.
Top: Flame type (Ia) at $\lambda_{eff} = 2.3$, $T_{mix} = 700[K]$, $\dot{m}_{air} = 40[g/s]$.
Middle: Flame type (Ib) at $\lambda_{eff} = 2.2$, $T_{mix} = 700[K]$, $\dot{m}_{air} = 40[g/s]$.
Bottom: Flame type (IIa) at $\lambda_{eff} = 1.8$, $T_{mix} = 700[K]$, $\dot{m}_{air} = 40[g/s]$.
Figure 5.5. Time-averaged images of the OH-PLIF intensity investigated in the long combustion chamber (LCC).

Left: Normalization of the intensity with the individual maximum of each image.
Right: Normalization of the intensity with the absolute maximum of all images.
Top: Flame type (Ia) at $\lambda_{afr} = 2.3$, $T_{mix} = 700 \, [K]$, $\dot{m}_{afr} = 40 \, [g/s]$.
Middle: Flame type (Ib) at $\lambda_{afr} = 2.2$, $T_{mix} = 700 \, [K]$, $\dot{m}_{afr} = 40 \, [g/s]$.
Bottom: Flame type (Irc) at $\lambda_{afr} = 1.8$, $T_{mix} = 700 \, [K]$, $\dot{m}_{afr} = 40 \, [g/s]$.
change of flame shape and height attends by a significant increase of sound emission takes place (the flame tip suddenly opens and extends into the outer recirculation zone; from flame types (Ib) to (IIa) and previously called “fast transition”). Secondly, the flame envelope is stretched in the streamwise direction and the flame seems to jump back and forth between two stationary states characterized by a longer and a shorter flame height (flame type (IIa)). This transitional flame exists for a narrow band of parameters, was previously referred to as the “slow transition” and is responsible for an increased pressure drop across the combustor as well as large pressure oscillations (major regime of thermoacoustic instabilities). By further decreasing $\lambda_{afr}$ and firing the combustor at richer operating conditions, the flame became continuously more compact and reached the strongly burning state of flame type (IIIa).

Since all investigated flame structures in the long combustion chamber are identically observed for the short combustion chamber, only the additionally investigated flame type (IIIb) of the SCC is given in Figure 5.6. A complete listing of all flame structures occurring in the SCC is appended in Chapter A.3.5. Although at a first glance the flame type (IIIb) seems to have a similar flame shape like flame type (IIa), flame characteristics are totally distinct. Flame type (IIIb) is compact and more or less stable (increasing high-frequency modes), whereby the characteristic of the flame structure (IIIb) is directly opposed in the sense that this flame type exhibits strong pressure fluctuations especially in the low-frequency range (See Figure 4.22 on page 71). This putative compact and stable-looking flame shape results from the time-averaged imaging technique blurring and smearing all of the flame movement. The similar effect is also valid for the unstable flame type (IIb) of the long combustion chamber. To clarify the differences between flame types (IIa) and (IIb) as well as to point out the own characteristic of these flame structures compared to all others, colour photos are taken at the end of the short com-

![Figure 5.6.](image)

**Figure 5.6.** Time-averaged images of the OH chemiluminescence intensity showing the additional flame type (IIIb) investigated in the short combustion chamber (SCC) at an operating condition of $\lambda_{afr} = 1.8$, $T_{mix} = 700{K}$ and $\dot{m}_{fuel} = 40{g/s}$. 

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bustion chamber showing still a well-developed reaction zone (See Figure 5.7). For all other flame structures, no combustion zone was observed so far downstream of the combustor (15 to 20 burner diameters).

The differences between OH-chemiluminescence and OH-PLIF images are visible and inherent to both acquisition techniques. The OH-chemiluminescence pictures result from an integration of emitted light along the optical path while OH-PLIF pictures are a representation of a vertical slice of the flame. Consequently, the measured chemiluminescence intensity is a combination of light emission and optical path thickness. For the same reason, the flame contours obtained with chemiluminescence are smoothed out and the signal-to-noise ratio was larger for PLIF pictures thus resulting in a better contrast. This effect is amplified by the relatively long exposure time of 400[μs] necessary for each OH-chemiluminescence single shot.

In addition, the single-shot measurement method allows further post processing compared to such techniques, where the ICCD chip is directly used to sum up the individual acquired single-shot images. By exporting every single image, expenses are increased dramatically and an enlarged memory capacity as well as a huge amount of disk space and an advanced post processing are required. In addition, extra time and effort spent are necessary for an accurate correction of the raw OH-PLIF images (correction of the laser sheet intensity distribution, of the laser shot-to-shot intensity variations of the different absorption losses, etc.) and justified for the chemiluminescence technique by an additional return. Both measurement techniques permit the derivation of the combustion probability and the probability of the flame front.

The post processing techniques needed to determine the flame front itself, to binary intensity images into burnt and unburnt zones and finally, to calculate the probability of combustion and flame front have already been described in Chapter 3.6.2 and are

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**Figure 5.7.** Color photos are taken at the atmospheric access at the upper end of the short combustion chamber (SCC). Left: Flame shape corresponding to flame type (1a). Right: Flame type (1b).
Figure 5.8. Probability of combustion and of the flame front occurrence derived from time-averaged images of the OH chemiluminescence and OH-PLIF intensity for a constant mass flow rate $m_{\text{air}} = 400 \text{g/s}$ and mixture temperature $T_{\text{mix}} = 700 \text{K}$.

From top down: Flame type (la) at $\lambda_{\text{eff}} = 2.8$, (lb) at $\lambda_{\text{eff}} = 2.6$, (la) at $\lambda_{\text{eff}} = 2.3$, (lb) at $\lambda_{\text{eff}} = 2.2$ and (lb) at $\lambda_{\text{eff}} = 1.8$.

therefore not mentioned again. Figure 5.8 shows a summary of the probabilities (combustion and flame front) derived from time-averaged OH chemiluminescence and OH-PLIF intensity images according to the images shown for flame types (la) - (lb) in the figures 5.2 up to 5.5. From a technical measurement point of view, the images showing the probability of the flame type (la) obtained from the OH chemiluminescence images are remarkable. Although the images are binarized, this flame types shows
surprisingly high values indicating a reaction zone outside of the typical small “V” shape (blue regime). If the same flame type is investigated with the OH-PLIF imaging technique, the combustion probability as well as the flame front probability are almost zero in the corresponding regions. Indeed, intensity images already having a very low signal-to-noise ratio (e.g. flame type 1a in Figure 5.2) comprise an increased uncertainty of the binarization process by the derivation of the probabilities resulting in weak representation of the probability of combustion and of the flame front.

Moreover, operating conditions exhibiting major thermoacoustic instabilities and thus having a naturally forced flame movement, own an increased region with the utmost probability of combustion (flame type 2b). Weak reaction zones, such as flame types 1a and 2a have only a small combustion probability since only the tip of the reaction zone is visible within the optical access of the test rig and most of the spread flame inside of the combustor is not investigable (see Figure 5.7). In contrary, strongly burning flames (flame type 3a) were stabilized closer to the burner exit and are even more compact resulting in a narrower area where combustion is able to take place.

Variation of the mixture temperature, air/fuel equivalence ratio and mass flow rate

According to the investigate operating conditions shown in Figure 5.1, time-averaged OH chemiluminescence and OH-PLIF intensity images are listed in a tabular form on the following pages. These tables are composed out of four columns, defining a fixed mixture temperature and seven rows, indicating a constant mixture strength each, whereby the air mass flow rate is kept constant for all investigated operating conditions of the tables.

Within the first two tables (Figure 5.9 and Figure 5.10), the normalization of the intensity values is calculated accordingly to the maximum of every single time-averaged image. Therefore, the colouring and the normalized intensity values respectively between the several depicted images could not be compared on a one-to-one basis. For instance, a red zone of a time-averaged intensity image acquired at a lean operating condition has not the identical intensity (absolute) value such as those red zones depicted for richer operating conditions.

Figure 5.11 shows images for which the normalization of intensity values is performed in the range defined by the absolute minimum and maximum intensity value found within all OH chemiluminesence images acquired, whereby Figure 5.12 shows the normalization within all OH-PLIF images respectively. The similar illustration of the influence of the mixture temperature and the air/fuel equivalence ratio of the flame structure and the corresponding OH chemiluminesence images obtained for the short combustion chamber were additionally listed in the appendix in Chapter A.3.5 (See Figure A.13 and Figure A.14 on page 177).
Flame structure

Figure 5.9: Time-averaged images of the OH chemiluminescence intensity for the long combustion chamber (LCC) according to Figure 5.1 ($m_{air} = 40\text{g/s}$).
Figure 5.10. Time-averaged images of the OH-PLIF intensity for the long combustion chamber (LCC) according to Figure 5.1 (\(\dot{m}_{\text{air}} = 400 \text{g/s}\)).
Figure 5.11. Time-averaged images of the OH chemiluminescence intensity (normalized over all images) for the LCC according to Figure 5.1 ($m_{air} = 40\text{g/s}$).
Figure 5.12. Time-averaged images of the OH-PLIF intensity (normalized over all images) for the LCC according to Figure 5.1 ($\dot{m}_{air} = 40 [g/s]$.}
To complete the investigated parameter space \((\lambda_{afr}, T_{mix} \text{ and } m_{air})\), time-averaged intensity images are shown in Figure 5.13 for an air mass flow variation within the range of \(m_{air} = [30, 36, 42, 48][g/\text{s}]\). The mixture temperature was set to \(T_{mix} = 700[K]\) and the air/fuel equivalence ratio to \(\lambda_{afr} = 2.2[-]\) and maintained at these levels during these measurements. To verify the results, an identical variation of the air mass flow rate was additionally performed by means of a distinct baseline operating condition \((T_{mix} = 600[K], \lambda_{afr} = 1.8[-]\) and is listed in the appendix (See Figure A.15).

A description of flame types and the influence of the parameters such as the mixture temperature, the mass flow rate and the equivalence ratio on the flame structure, apart the already stated outcomes, remains difficult to interpret the previously shown inten-

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Figure 5.13. Time-averaged images of the OH chemiluminescence and the OH-PLIF intensity whereby in the second and the fourth columns the images listed are normalized within all images) for the LCC. Operating conditions: \(\lambda_{afr} = 2.2, T_{mix} = 700[K]\) and \(m_{air} = [30, 36, 42, 48][g/\text{s}]\) from the bottom up.
Flame structure

For this reason, the flame front position and the light intensity center of gravity have been taken into consideration for an improved understanding and the ability to qualify the influence of every single parameter such as $\lambda_{afr}$, $T_{mix}$ and $\dot{m}_{air}$. Along the next paragraphs, these influences are consequentially discussed.

### 5.1.2. Influence of the air/fuel equivalence ratio

This section focuses on the influence of air/fuel equivalence ratio variations on the averaged flame front position and flame center of gravity for a fixed air mass flow rate of $\dot{m}_{air} = 40\,[g/s]$ and two distinct mixture temperatures of $T_{mix} = 600\,[K]$ and $T_{mix} = 700\,[K]$. The air/fuel equivalence ratio was varied by steps of $\Delta\lambda_{afr} = 0.2$ keeping the air mass flow rate at a constant level. The flame front location measured with both techniques is shown in Figure 5.14 for $\lambda_{afr} = 1.8, 2.0, 2.2, 2.4, 2.6$ and $2.8$.

![OH-Chemiluminescence](image1)

![OH-PLIF](image2)

**Figure 5.14.** Flame front position for a constant mass flow rate ($\dot{m}_{air} = 40\,[g/s]$) and a mixture temperature of $T_{mix} = 600\,[K]$ (top row) and $T_{mix} = 700\,[K]$ (lower row) for several air/fuel equivalence ratios, $\lambda_{afr} = 1.8, 2.0, 2.2, 2.4, 2.6, 2.8\,[\cdot]$. (flame is extinguished for $\lambda_{afr} = 2.8\,[\cdot]$ and $T_{mix} = 600\,[K]$)
Figure 5.14 shows that, for decreasing values of the air/fuel ratio (richer operating conditions), the flame front continuously moved towards the burner outlet and the flame tip became simultaneously wider. Between $\lambda_{afr} = 2.0[-]$ and $\lambda_{afr} = 2.2[-]$ for $T_{\text{mix}} = 600[K]$ and between $\lambda_{afr} = 2.2[-]$ and $\lambda_{afr} = 2.4[-]$ for $T_{\text{mix}} = 700[K]$, the average flame front location and shape suddenly changed. Below these critical values, the flame stabilized significantly closer to the burner and the flame front changed from a “V” shape to a “M” shape with the flame edges at the same height as the flame center. In addition, the flame border was extending into the outer recirculation zone. The flame front formed two more or less pronounced arches coinciding with the burner outlet walls at $\pm 25[mm]$ where the axial velocities were maximum. This rapid and significant variation of the flame front shape corresponds to the fast transition previously described and tends to confirm the existence of two distinct stabilizing mechanisms associated with two limit cycles. The critical air/fuel equivalence ratios obtained for this transition from chemiluminescence and PLIF images are equal and in good agreement with acoustic measurements for both mixture temperatures. Furthermore, the average flame front height measured with both optical diagnostics is in good agreement.

However, since chemiluminescence measurements are the result of a projection of a 3-dimensional flame onto the plane of the camera’s ICCD chip, the real flame front shape revealed by PLIF measurements was averaged out by optical integration. The comparison between both mixture temperatures shows that, for equal mixtures concentrations, the flame stabilized closer to the burner outlet for the hotter mixture, as
expected, and the transition occurred for a leaner air/fuel ratio. Finally, chemiluminescence and PLIF measurements for $T_{\text{mix}} = 700[K]$ both indicate that the flame front location and shape were hardly affected by $\lambda_{afr}$ once the fast transition took place, unlike for the lower mixture temperature.

The center of gravity of the light intensity (CoG) was calculated individually for the four regions ROI1-ROI3 defined in Figure 3.15 and shown in Figure 5.15. The nearly constant location of the center of gravity, especially for the left and right regions of interest (ROI1 and ROI2) for air/fuel equivalence ratios leaner than $\lambda_{afr} = 2.2$ (after the fast transition) is due to the poor signal-to-noise ratio of chemiluminescence. The difference between flame and background signal is relatively small for the OH chemiluminescence technique applied to these lean operating conditions and the center of gravity is therefore less affected by the reaction zone. In particular, the CoG for ROI1 and ROI2 possess no tendency to move towards the axis of symmetry of the combustor for operating conditions with $\lambda_{afr} \geq 2.4$ although the time-averaged intensity images indicate a flame stabilized in a distant region close to the axis of symmetry (see Figure 5.2).

In counterpart, the PLIF signal-to-noise is much better (the exposure time is so small that the background light is literally zero) so that the CoG correctly reflects the flame behaviour even when the relative flame surface is small. In addition, the CoG locations measured with PLIF are globally higher above the burner outlet due to optical integration of chemiluminescence resulting in a compact flame envelope as visible in the combustion probability images of Figure 5.8. Especially the “M” shape of the flame front is more accentuated for PLIF and incites the CoG to relocate its center towards the end of the combustion chamber.

However, the discontinuity previously observed for flame fronts due to the fast transition is clearly visible in the CoG measurements for both chemiluminescence and PLIF. It confirms that the entire flame has been translated upstream against the flow as the transition occurred.

5.1.3. Influence of the mixture temperature

The same investigation was repeated by varying the mixture temperatures for a constant air mass flow rate of $m_{\text{air}} = 40[g/s]$ and constant air-to-fuel ratios of $\lambda_{afr} = 1.8[-]$ and $\lambda_{afr} = 2.2[-]$. The results for mixture temperatures of $T_{\text{mix}} = 600, 650, 700$ and $750[K]$ for the long combustion chamber are shown in Figure 5.16.

The average flame front position measured with both techniques was identical and shows the same trends. Unlike for air-to-fuel ratio variations, for which a smooth and continuous evolution was observed before and after the fast transition took place, the
Figure 5.16. Flame front position for a constant mass flow rate ($m_{air} = 40[g/s]$) and an air/fuel equivalence ratios of $\lambda_{afr} = 2.2[-]$ (top row) and $\lambda_{afr} = 1.8[-]$ (lower row) for several mixture temperatures $T_{mix} = 600, 650, 700, 750[K]$ (LCC).

variations of mixture temperature seem to exhibit a strongly binary behaviour. By making a rough estimation, the change of energy introduced into the system by increasing the mixture temperature of $\Delta T_{mix} = 50[K]$ is in the same order of magnitude as lowering the air/fuel equivalence ratio of $\Delta \lambda_{afr} = 0.2$ ($\Delta T_{mix} = 50[K] \leq \Delta T_{f,ad} = 75[K]$ and $\Delta T_{f,ad} = 75[K] = \Delta \lambda_{afr} = 0.2$). Therefore, the stronger binary behaviour of the flame fronts is probably due to the wider discretisation of the investigated temperature steps compared to the air/fuel equivalence ratio measurements.

The flame front location as well as the shape for $\lambda_{afr} = 2.2[-]$ and both colder temperatures was identical showing no or small influence of $T_{mix}$. The same observation holds for the two hotter mixtures once the fast transition took place (see CoG in Figure 5.17).

This behaviour is confirmed by the measurements for $\lambda_{afr} = 1.8[-]$ where almost no effect due to mixture temperature is visible on the flame front location. Note that for this relatively rich mixture the flame was in “a strongly burning” state for all investi-
Figure 5.17. Light intensity center of gravity for ROI0 to ROI3 for the investigated operating conditions with the LCC: \( m_{air} = 40 \text{[g/s]} \), \( T_{min} = [600, 650, 700, 750]\text{[K]} \) and \( \lambda_{air} = [1.8, 2.0, 2.2, 2.4] \) from the bottom up to the images on the top row.
gated temperatures and thus the fast transition is not visible in Figure 5.16 (as well as for the CoG images depicted in Figure 5.17). This binary behaviour constitutes additional evidence supporting the existence of two stabilizing mechanisms associated with two distinct limit cycles. Additionally, the mixture temperature has an influence on the occurrence of thermoacoustic instabilities quite similar to the effects of a changing mass flow rate and this behaviour is described therefore in the next section in more details.

The small dependency of the flame front location on the mixture temperature is confirmed by the measurements of the light intensity center of gravity shown in Figure 5.17 for the left, middle, right and entire regions of the visualization. Although a small effect of mixture temperature is visible (apart the sudden transition), in particular for the leanest case studied, the variation stayed limited. Since the flame front remains more or less at its position for rich and hot conditions and the CoG exhibits still a small tendency to move towards the burner outlet, the overall flame shape and

Figure 5.18. Position of the flame front for distinct air mass flow rates (LCC):
Top row: $T_{\text{mix}} = 700\,[K], \lambda_{\text{air}} = 2.2$ and $m_{\text{air}} = [30, 36, 42, 48]\,[g/s]$.
Lower row: $T_{\text{mix}} = 600\,[K], \lambda_{\text{air}} = 1.8$ and $m_{\text{air}} = [30, 36, 42, 48]\,[g/s]$. 
therefore the reaction zone as well, must be more condensed due to an increased mixture temperature and a richer air/fuel equivalence ratio respectively.

5.1.4. Influence of the mass flow rate

The air mass flow rate modifies mainly the thermal power density in the combustion chamber and hydrodynamics at the burner outlet by increasing velocities. This latter effect impacts the strength of the recirculation zone improving the mechanism of flame stabilization for larger mass flow rates. This trend is clearly visible in Figure 5.18 (top row) where the flame front location is shown for $T_{\text{mix}} = 700[K]$, $\lambda_{\text{air}} = 2.2[-]$ and for mass flow rates of $m_{\text{air}} = [30, 36, 42, 48][g/s]$. Although the flame front continuously moved towards the burner outlet as $m_{\text{air}}$ was increased, a sudden change of flame shape was observed between $m_{\text{air}} = 36$ and $m_{\text{air}} = 42[g/s]$ along with the transitions of flame type $\Xi$ to $\Theta$. For $m_{\text{air}} = 42[g/s]$, the mixing rate associated with the outer recirculation zone reached a critical value enabling the stabilization of the flame close to the burner exit (see PLIF measurements). This behaviour is confirmed for the colder but richer mixture shown in Figure 5.18 (lower row).

Although, increasing the burner exit velocity for these operating conditions moved the averaged flame front position away from the burner (see chemiluminescence data) with a maximum reached in the region of large axial velocities (approximately from ±15 to ±27[mm]) (see PLIF measurements), the unstable flame type $\Xi$, having the flame front closest to the burner exit (only for time-averaged measurements!), is transferred to the more stable flame type $\Theta$ by this rise of the mass flow rate. The mixture velocity has thus apparently two effects; for leaner flame types ($\Xi$, $\Theta$ and $\Xi$) the dominant parameter governing flame stabilization seems to be mixing between hot combustion gases and the fresh mixture in the recirculation zone. This improved mixing rate for higher mass flow rates makes the flame types tight and strongly fixed at the burner exit (see Figure 5.13).

For the unstable flame type $\Xi$, the hydrodynamic time scale seems to affect the characteristic time of the feedback process responsible for the appearance of thermoacoustic instabilities. For instance, the altered mixing rate changes the characteristic time of heat release fluctuations which may result in a less effective coupling with the acoustic properties of the combustion chamber (for the corresponding chemiluminescence and LIF images see Figure A.15 in the appendix on page 179). However, an increasing mass flow rate results not strictly in a less effective coupling, as this example shows for the long combustion chamber. The risen pressure oscillations for elevated mass flow rates within the short combustion chamber are evidence of shifted time scales (e.g. fluid dynamic time scales) inducing a more sensitive coupling to force thermoacoustic instabilities (flame type $\Xi$).
The trends previously described for the flame front are confirmed by the behaviour of the flame center of gravity as a function of mass flow rate, in particular for the leaner mixture (see Figure 5.19, top row). For the richer condition (see Figure 5.19, lower row) the motion of the center of gravity with the mass flow rate is so small that it can neither contradict nor confirm the observations made on the flame front location.
5.2. Unsteady flame dynamics and the flame structure

5.2.1. Dynamic characterization of the different flame types

Phase-averaged images were captured in order to visualize the cyclic response of the flame over one time period of a forcing signal or of a natural frequency of the investigated system. The advantage of using a forcing signal is obvious for the phase-locked measurement technique itself. Due to the synthetically generated signal, the resultant forcing signal can be used for a perfect triggering scheme of the ICCD cameras and laser system. In contrary, the drawback of forcing a system (e.g. by a loudspeaker or a fuel modulation, etc.) is the inherent modification of the characteristics of the flame and the response of the flame remains always forced, thus is not the natural behaviour of the flame itself.

In particular with regard to thermoacoustic instabilities, where the coupling between the acoustic and flow field as well as the heat release rate are investigated, a forcing of one of these parameters may interfere with the others and may affect the achieved results adversely. A forcing of one parameter must not necessarily result in a resonant coupling of all involved parameters. Hence, it was abstained from an active forcing of the flame structure for all phase-averaged measurements of the current work to exclude the hazard of measuring a “synthetic” flame along with the corresponding forcing mechanism and having the uncertainty of investigating suppressed or even no natural qualities.

Restricted to naturally occurring signals, more care has to be taken regarding the use of the reference signal and its quality. Several tests have been carried out to obtain the most suitable reference signal for the triggering signal. All possible locations of the microphones as well as the OH photomultiplier signal came into consideration, whereby the best results were achieved by using MIC2 (microphone closest to the reaction zone, see Figure 3.3) as the reference signal due to its best signal-to-noise ratio and excellent pressure response behaviour for all investigated flame types.

Though the signal may have a significant peak in the power spectrum and a specific frequency is dominant within this frequency domain, an appropriate triggering signal is not already warranted. The dominant frequency content must be easily detectable in the time domain both to ensure a correct phasing of the imaging technique and to obtain an adequate phase-averaged image (rather than a misconducted time-averaged image). Even if the triggering system is technically applicable to such cases, it makes no sense to attempt to measure flame types where no dominant frequency is present in the time domain of the signal too. Therefore, the flame type \( \text{Flame Type A} \) has not been investigated with the phase-averaged imaging technique due to the complete absence of such dominant frequency content.
Flame type \( \text{\textit{a}} \)

For flame type \( \text{\textit{a}} \), a set of phase-averaged images has been acquired even though a dominant frequency is marginally present. A cyclic variation of the OH chemiluminescence and the OH-PLIF intensity is not visible from Figure 5.20. The additionally investigated positions of the flame front during one time period, shown in Figure 5.21 remain almost at a constant location and exhibit no cyclic variation. This provides a further evidence of the justified classification as a stable flame previously derived with the acoustic characterisation of this flame type (see Chapter 4.4).

Flame type \( \text{\textit{a}} \)

Figure 5.22 shows phase-averaged intensity images taken at an operating conditions laying in the region of the transition line between flame type \( \text{\textit{a}} \) and \( \text{\textit{b}} \). As can be seen from the OH chemiluminescence figures, the intensity values itself as well as the

![Figure 5.20](image-url)

Figure 5.20. Phase-averaged images of the OH chemiluminescence and OH-PLIF intensity (normalized over the phase) for a constant mass flow rate \( m_f = 40 \text{g/s} \), a mixture temperature of \( T_{\text{mix}} = 700 \text{K} \) and an air/fuel equivalence ratio of \( \lambda = 2.4 \) (LCC).
flame shape is cyclic altered for each single image. The reaction zone is slightly growing and shrinking according to the pressure signal (triggering reference). In contrary to the chemiluminescence images, this effect is less visible to the naked eye within the phase-averaged OH-PLIF images. However, the several positions of the flame front inferred from the OH-PLIF intensity images during one time period of the dominant frequency (at 202 [Hz] for this specific operating condition) indicate a movement of the flame as well. From Figure 5.23 it can be seen that the cyclic displacement of the flame front has the similar order of magnitude for chemiluminescence as for PLIF.

The two arches of the “M”-shape of the flame front have already been identified to be highly sensitive and its reaction of a changing air/fuel equivalence ratio was described in details in section 5.1.2 et seq. (The mixture temperature and the mass flow rate also show a minor influence on the arches). In consideration this fact, it is not surprising that these curvatures are warped to different degrees during one period of instability.

Figure 5.21. Position of the flame front (top row) and velocity profile of the flame front evaluated in the y-axis direction (lower row) for a constant operating condition of the LCC ($h_{air} = 40 [\text{g/l}], T_{mix} = 700 [\text{K}]$ and $\lambda_{off} = 2.4$).
mainly within this region. Contrary to the arches of the “M”-shape, the position of the flame front in the area at the axis of symmetry of the combustion chamber (the central point of the “M”-shape) is less affected and remains almost stable for all phasings.

By knowing the frequency of instability driving the movement of the flame front, the phase angle during which the phase-averaged image was exposed and the displacement of the flame front within two consecutive phase angles, a velocity of the flame front motion itself is deducible by assuming a strict propagation of the flame along the y-axis of the combustor (e.g. see the lower row of Figure 5.23). The absolute values of the calculated velocities should be treated with care due to the assumption of a unidirectional flame propagation and do not represent a reaction rate at all.

\[ \text{Figure 5.22. Phase-averaged images of the OH chemiluminescence and OH-PLIF intensity} \]

(normalized over the phase) for a constant mass flow rate \( m_{\text{in}} = 40 [g/s] \), a mixture temperature of \( T_{\text{mix}} = 700[K] \) and an air/fuel equivalence ratio of \( \lambda_{\text{eff}} = 2.2 \) (LCC).
Flame type (IIb)

Flame type (IIb) (and for the SCC flame type (IIIb)) was previously mentioned especially in conjunction with large amplitudes of the pressure oscillation within the combustor, an increased pressure drop across the combustor as well as with major fluctuations of the OH radical obtained by a photomultiplier tube (PMT) close to the burner outlet. The impact of the thermoacoustic instability (dominant frequency at $223\,[Hz]$) on the flame shape is depicted with the phase-averaged images of Figure 5.24. The OH chemiluminescence and the OH-PLIF images show a strong variation of the intensity values associated with an altered flame shape. The flame structure derived from the chemiluminescence images technique is characterized by an alternating behaviour from a weak and compact flame shape to an intensely burning and widely spread reaction zone. As can be seen from the PLIF images, the two streams of the fresh mixture entering the combustion chamber are changing the penetration depth and its

![OH-Chemiluminescence](image1)

![OH-PLIF](image2)

**Figure 5.23.** Position of the flame front (top row) and velocity profile of the flame front evaluated in the y-axis direction (lower row) for a constant operating condition of the LCC ($m_{air} = 40\,[g/s]$, $T_{air} = 700\,[K]$ and $\lambda_{eff} = 2.2$).
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Angular direction. The OH-PLIF image obtained at a phase angle of $\phi_{trig} = 0^\circ$ exhibits a reaction zone like a stout cork trying to clog the burner. In this case, the penetration of the fresh mixture into the reaction zone is small and the azimuthal swirl flow of the fresh mixture is located closer at the walls of the combustor. After 135 degrees, the fresh mixture enters deep into the reaction zone. In return, the inner recirculation zone of the reactive flow is thin and fragile, but still reaches the dump plane of the combustor. During the next time steps, the reaction zone is recovered and the “cork” is again fully established till the end of the instability period. This behaviour of the movement of the flame front reinforces the explanation of an increased pressure drop across the burner for the unstable flame structure of flame type (iii) already stated in Chapter 4.4.

The variant penetration of the fresh mixture into the reaction zone strongly influences the location of the flame front, whereby the arches of the “M”-shape of the flame front are stretched by the movement of the flame. For the flame front obtained from the

**Figure 5.24.** Phase-averaged images of the OH chemiluminescence and OH-PLIF intensity (normalized over the phase) for a constant mass flow rate $\dot{m}_{q,c} = 40$ g/s, a mixture temperature of $T_{mix} = 650[K]$ and an air/fuel equivalence ratio of $\lambda_{efr} = 2.0$ (I.C.C).
chemiluminescence images, the “M”-shape disappears almost at a phase angle of 45° and is fully redeveloped 180° later. For both techniques, the central point of the “M”-shape is less affected and remains for all phasings at the identical position. The symmetric formation and movement of the flame front as well as the complete reaction zones by showing a symmetric varying intensity (chemiluminescence and PLIF) for the different phase angle, indicate a heat release rate dominantly forcing the longitudinal acoustic mode(s) within the combustion chamber.

Flame type

Since no pressure oscillation at a single frequency is dominating the flame type Q1, an adequate triggering is difficult. The pressure signal used for the triggering scheme mainly consists out of two equipollent signals at distinct frequencies (259[Hz] and 360[Hz]) and out of a high-frequency signal with smaller amplitudes oscillating at 4.7[kHz]. Since the high-frequency content is hardly noticeable at this level in the time

Figure 5.25. Position of the flame front (top row) and velocity profile of the flame front evaluated in the y-axis direction (lower row) for a constant operating condition of the LCC ($\dot{m}_{air} = 40[g/s]$, $T_{mix} = 650[K]$ and $\lambda_{eff} = 2.0$).
domain, this signal was not feasible for the use of a timing signal. Therefore, the triggering setup has been individually tuned for both of the lower frequencies (even this introduces an increased uncertainty compared to the dominant frequency of flame type (fib)).

The phase-averaged intensity images as well as the flame front position and the calculated velocities of the flame front do not mark out a particular difference in the flame characteristics. Hence, it has been renounced to list both sets of acquired images at this position. The images shown in Figures 5.26 and 5.27 were obtained with the triggering scheme optimised for the lowest frequency. The intensity of the chemiluminescence and PLIF images are only marginally changed during one time period of the chosen triggering frequency. A displacement of the flame front is only identifiable for the “M”-shaped flame fronts acquired with the OH-PLIF technique. The center of the “M”-shape of the flame front lurches around the axis of symmetry of the combustor like a

![OH-Chemiluminescence and OH-PLIF Images](image)

**Figure 5.26.** Phase-averaged images of the OH chemiluminescence and OH-PLIF intensity (normalized over the phase) for a constant mass flow rate $m_{\text{in}} = 40 \text{[g/s]}$, a mixture temperature of $T_{\text{mixture}} = 700 \text{[K]}$ and an air/fuel equivalence ratio of $\lambda_{\text{air}} = 1.8$ (LCC).
spinner. While the center is going round, the left arch resulting from the penetrating fresh mixture is even more stretched and the other is bending closer to the burner exit. This results in y-velocity components in an opposite direction. Although these velocities have to be judged with caution, one may reasonably expect the presence of an azimuthal mode, which corroborates the statements in section 4.4.2. However, a clear identification of the mode order is not possible from these phase-averaged images. The acoustic identification of the combustor derived with a finite element code (see Chapter 4.1) assigns a first azimuthal mode along with the 4.7[kHz] frequency dominating the high frequency range of the spectrum obtained for the current operating condition.

Flame type (SCC only)

A near relative to the unstable flame type of the long combustion chamber is the flame type (SCC) investigated in the short combustion chamber (SCC). Besides the obvi-

![Figure 5.27. Position of the flame front (top row) and velocity profile of the flame front evaluated in the y-axis direction (lower row) for a constant operating condition of the LCC \( \dot{m}_{\text{air}} = 40\, \text{[g/s]}, \ T_{\text{mix}} = 700\, \text{[K]} \) and \( \lambda_{\text{mix}} = 1.8 \).]
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Figure 5.28. Phase-averaged images of the OH chemiluminescence intensity (normalized over the phase) for a constant mass flow rate $\dot{m}_{\text{air}} = 40$ g/s, a mixture temperature of $T_{\text{mix}} = 750 [\text{K}]$ and an air/fuel equivalence ratio of $\lambda_{\text{air}} = 1.8$ (SCC). Owing to the similarities, the flame shape is generally more compact at the moment of the highest chemiluminescence intensity compared to the intensity of flame type (Iib). This effect can be seen for operating conditions representing the same level of pressure oscillations for both long and short combustion chamber. Figure 5.28 shows phase-averaged images acquired for the SCC and Figure 5.30 images taken with the LCC. Since thermoacoustic instabilities appear for different operating conditions for the SCC and LCC, the air/fuel equivalence ratio, the mixture temperature as well as the mass flow rate were variably set to obtain a sound pressure level (SPL of MIC2) which matches both combustion chambers. The flame front itself is characterized by a less pronounced “M”-shape. During a period of the dominant instability (at a frequency of 541 [Hz]),

Figure 5.29. Position of the flame front (left) and velocity profile of the flame front evaluated in the y-axis direction (right) for a constant operating condition of the SCC ($\dot{m}_{\text{air}} = 40$ g/s), $T_{\text{mix}} = 750 [\text{K}]$ and $\lambda_{\text{air}} = 1.8$.)
the two arches are less pronounced and even located closer at the exit of the burner as for the stabler flame type \( \text{(a)} \). Although the tendency is even present for flames of type \( \text{(b)} \), flame oscillations shift the midpoint of the “M”-shape stronger than the arches. This fact is indicated by the horizontal blue line for a phase angle of 90° in Figure 5.29. The horizontal line is the result of the optical limit by the flame detection due to the penetration of the flame into the EV burner. In defiance of a flame type stabilizing close at the dump plane and a flame intermittently penetrating the combustor, the increased wall temperature poses no problem for the combustor (e.g., no glowing, etc.).

5.2.2. Combustion and flame front probability

Corresponding to the chemiluminescence and LIF intensity images of Figure 5.30 showing (in terms of pressure oscillations) the most unstable operating conditions investigated in the long combustion chamber, the phase-average combustion probabil-

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**Figure 5.30.** Phase-averaged images of the OH chemiluminescence and OH-PLIF intensity (normalized over the phase) for a constant mass flow rate \( \dot{m}_{\text{air}} = 30 \text{g/s} \), a mixture temperature of \( T_{\text{mix}} = 600 \text{ K} \), and an air/fuel equivalence ratio of \( \lambda_{\text{air}} = 1.8 \) (1:CC).
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ity during one period of instability has been derived (see Figure 5.31). The phase-averaged probability of the location of the flame front has additionally been calculated and is listed as additional information in the appendix on page 180 (Figure A.16).

As can be seen from Figure 5.31, the reaction zone is cyclically swaying from a weak (small region) to a strong reaction zone with a large region of a high combustion probability driven by major thermoacoustic instability. For the two-dimensional PLIF technique, it seems like a “cork” is trying to obturate the exit of the EV burner (especially at the phase angles $\phi_{\text{trig}} = 315^\circ$, $0^\circ$ and $45^\circ$). If the fresh mixture attempting to enter the combustion chamber prevails ($\phi_{\text{trig}} = 90^\circ$), a vortex of fresh mixture sheds at the burner exit (where a large axial velocity component is present) and penetrates deep into the reaction zone. The overall probability of combustion degrades ($\phi_{\text{trig}} = 225^\circ$ and $270^\circ$) and finally, the reaction zone branches backward and a “cork” is established afresh.

**Figure 5.31.** Probability of combustion derived from phase-averaged OH chemiluminescence and OH-PLIF intensity images for an operating condition of $\phi_{\text{trig}} = 1.8$, $T_{\text{mix}} = 600[K]$ and $\dot{m}_{\text{air}} = 30/[\text{g/s}]$ (LCC).
Furthermore, the time period during where the "cork" is dominating, corresponds to such images derived from chemiluminescence, which shows large regions of a high combustion probability. Besides, the time periods where the fresh mixture is mainly penetrating the reaction zone (vortex shedding) coincides with the combustion probability images of chemiluminescence exhibiting small reaction zones close to the burner. To put it in another way, the large regions of combustion probability deducted from chemiluminescence are almost in phase related to those of the combustion probability calculated from the PLIF images. Naturally, this is an astonishing outcome since the flame shape is not rotational symmetric due to the asymmetric design of the EV burner and, as a consequence, the vortices itself are generated at the slots of the EV burner only.

In addition, the last section of this chapter examines systematically the flame dynamics (light intensity and flame front position) between the two measurement techniques for all investigated operating conditions according to Figure 5.1.

5.2.3. Phase-averaged light intensity center of gravity

The center of gravity method calculates the center of the light intensity and therefore represents the gravity of the reaction zone within a specific region of interest as well. By a straightforward analysis of the phase-averaged intensity images, the center of gravity allows the visualization of the movement of the reaction zone during a complete time period of the instability.

Accordingly the center of gravity has been derived from the phase-averaged chemiluminescence images for the long combustion chamber for three different operating conditions, each representing a typical flame type. Figure 5.32 shows the center of gravity

![Figure 5.32. Center of gravity for ROI1 (left) and ROI2 (right) of the OH chemiluminescence intensity images for the long combustion chamber (LCC) at a constant mixture temperature of $T_{\text{mix}} = 650[K]$, a mass low rate $m_{\text{air}} = 40[g/s]$ and at three different air/fuel equivalence ratios $\lambda_{afr} = [1.8, 2.0, 2.2]$[-1].](image-url)
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for the flame type at the (iii)/ (iiia) transition without any noticeable variation of its position \( \lambda_{afr} = 2.2 \). In general, the mean position of the gravity center of this flame type is shifted closer to the combustor exit due to the already identified influences of a richer air/fuel equivalence ratio, a higher mixture temperature and an increased mass flow rate (for details see sections 5.1.2 up to 5.1.4). By doing so, the dynamic movement of the CoG for ROI1 as well as for ROI2 is imperceptibly enlarged during the time period of the dominant pressure oscillation (a few millimetres). There are, as one might expect, a large visible movement of the center of gravity along with the unstable flame type (iii). This dislocation, mainly in the y-axis direction, is enlarged and reduced according to the pressure oscillation (e.g. see rms sound pressure levels depicted in Figure 4.11) and at the end of this slow transition, the strong flame of flame type (iii) is established. Since the stable flame type (iiia) possesses smaller pressure amplitudes, the dynamical movement of the gravity center is reduced to a distance of a few millimetres.

Particularly, the track of the gravity center for flame type (iii) consists of parts located closer at the burner exit than the resulting curves of flame type (iiia). Similarly, such an effect was already observed by the location of the flame front derived from time-averaged images. In fact the dynamical movement of the unstable flame type (iii) results in a time-averaged flame front closer to the burner exit than all other flame types (e.g. see section 5.1.4).

Figure 5.33 shows the dynamic movement of flame type (iii) of the long combustion chamber (LCC) in conjunction with the unstable flame type (iii) investigated in the short combustion chamber (SCC). Both lanes of the gravity center for the LCC and SCC show the main alignments along the y-axis and only an inferior motion in the

![Figure 5.33. Center of gravity for ROI1 (left) and ROI2 (right) of the OH chemiluminescence intensity images (normalized over the phase) for the long combustion chamber (LCC) at \( \lambda_{afr} = 1.8 \), \( T_{mix} = 600\text{[K]} \), \( m_{air} = 30\text{[g/s]} \) and the short combustion chamber (SCC) at \( \lambda_{afr} = 1.8 \), \( T_{mix} = 750\text{[K]} \) and \( m_{air} = 40\text{[g/s]} \).]
direction coinciding with dump plane. The depicted operating conditions hold a comparably overall sound pressure level (rms SPL of MIC2), whereby the combustion chambers features pressure oscillations at distinct dominant frequencies of 215 [Hz] (LCC) and 541 [Hz] (SCC). By considering the distance covered by the moving center of gravity along the track and taking the corresponding dominant frequency of the pressure oscillation into account, a constant velocity of propagation of 7.5 [m/s] is resulting for the long combustion chamber and of 13.5 [m/s] for the short combustion chamber. Moreover, the geometry factor of 2, introduced by the aspect ratio of the LCC and SCC fits better to these calculated velocities than to the ratio derived from the dominant frequencies obtained for the different combustion chamber lengths (1.8 to 2.5).

Undoubtedly, this result is unexpected since the geometry, especially the combustion chamber length, is directly coupled with acoustic oscillations. The flame response, therefore the movement of the center of gravity as well, is lagging behind acoustic oscillations regarding the responsible feedback process characterising thermoacoustic instabilities. In conclusion, the flame structure can be characterized by having identical properties like flame type, whereby the acoustics is shifted to higher frequencies and the flame concurrently exhibits a faster response.

In addition, Figure 5.34 compares the movement of the gravity center during one time period obtained from OH chemiluminescence and OH-PLIF intensity images for the region of interest ROI1 and ROI2 (the original intensity images are shown in Figure 5.30). There is, as one might expect, a major difference between the tracks resulting from the dynamical movement of the CoG. The center of gravity of the PLIF images show a pronounced oscillation in the direction of the dump plane and compared to chemiluminescence, a rather negligible movement along the y-axis. Apart

Figure 5.34. Center of gravity for ROI1 (left) and ROI2 (right) of the OH chemiluminescence and the OH-LIF intensity images for the long combustion chamber (LCC) at $\lambda_{air} = 1.8$, $T_{mix} = 600\,\text{K}$ and $m_{air} = 30\,\text{[g/s]}$. 

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from this, the direction of rotation of the CoG motion has to be emphasised since it is in reverse direction to the flow field of the inner recirculation zone of the EV combustor.

If the region of interest ROI1 and ROI2 would have been set differently, the center of gravity movement received from the PLIF images will exhibit a more accentuated oscillation along the y-axis. In other words, the deep penetrating vortices of the fresh mixture and the slightly alternating reaction zone along the axis of symmetry and at the combustor walls were summed-up in such a way for the ROI1 and ROI2 that the resulting center of gravity is less affected in the y-axis direction. Indeed, if the region of interest is set for a region showing basically vortices, the motion of the CoG is developed in a similar manner in the longitudinal direction as for the chemiluminescence. However, in order to adequately compare several results of distinct operating conditions and in no case to “put the cart before the horse”, the region of interest has not been adapted to show unfavourable tuned similarities.

Figure 5.35. Averaged flame front positions of the region of interest ROI1 and ROI2 (top row) and the corresponding result of the sine curve fitting within all investigated ROI of the phase-averaged OH chemiluminescence and OH-PLIF intensity images for an operating condition (LCC) of $\phi_{air} = 2.0$, $T_{mix} = 650[K]$ and $m_{air} = 40[g/s]$. 
5.2.4. Generalization of the dynamic flame response

As a consequence of the various operating conditions investigated according to Figure 5.1 with the phase-averaged imaging technique, a further description of the dynamic flame response is needed to derive from the colourful intensity images beside the classification and assignment of typical flame fronts as well as its shapes to the distinct flame types \( \text{(in) to (ii)} \). The previous chapters have shown an extraordinary sensitivity for the position of the flame stabilization as well as a strongly varying chemiluminescence (and induced fluorescence) intensity variation along with changing operating conditions. Consequently, the position of the reaction zone relative to the EV burner as well as the actual heat release rate is a matter of particular interest to characterize the dynamic flame response.

To this end, two “lumped” parameters were derived to afford a narrow description during one time period of instability and to couple the dynamic flame response to acoustic pressure oscillations present in the combustion chamber. First, the parameter \( (I_{\text{norm}}, \text{ROI0-3}) \) representing the dynamic response of an overall reaction rate has been estimated as an averaged value obtained from the normalised OH chemiluminescence (and OH-PLIF) intensity within the specific regions of interest (ROI0-ROI3, defined according to Figure 3.15). Second, the position of the reaction zone has been described with an averaged parameter (indicated as Position \( y \)) showing the mean location of the flame front position observed in the particular region of interest (ROI0-ROI3).

The graphs of the images depicted in the upper row of Figures 5.35 and 5.36 show the averaged flame front position and the mean intensity respectively for an exemplary operating condition of \( \lambda_{\text{afr}} = 2.0, T_{\text{mix}} = 650[\text{K}] \) and \( \dot{m}_{\text{air}} = 40[\text{g/s}] \) during a period. It goes without saying that the development of these values (Position \( y \) and \( I_{\text{norm}}, \text{ROI0-3} \)) during a period resembles a sinusoidal curve, since the flame response is coupled with the sinusoidal pressure oscillations. (The black curve within these two figures indicates the phasing of the dominant pressure wave). Accordingly, these values derived from the intensity images at different phase angles were straightforwardly approximated with a sine function. The parameters of this function given in equation 5.1 (a universal formulation of a sine wave) were identified using a least-squared algorithm, whereby the amplitude \( A_{\text{sfit}} \), the offset \( B_{\text{sfit}} \) and the additional phase angle \( \phi_{\text{sfit}} \) had to be determined. The angular frequency \( \omega \) is already defined by the dominant pressure oscillation used as the reference signal for the imaging (trigger signal) and, as a consequence, is set equal to \( 2\pi f_{\text{trig}} \).

\[
f(t) = A_{sfit} \cdot \sin(\omega t + \phi_{sfit}) + B_{sfit}, \text{ where } -\pi \leq \phi_{sfit} < \pi
\]  

(5.1)
By using this definition of a sine wave, a negative phase angle $\phi_{\text{fit}}$ corresponds to an additional time lag with respect to the phasing of the pressure wave, whereas positive phase angles shift the resulting curve prior to the pressure signal and vice versa.

The results achieved from the fitting process are depicted exemplarily in graphs on the top rows of Figures 5.35 and 5.36 for the ROI1 and ROI2. On the lower row of these images, the graphs show the results achieved for all regions of interest. Although all conditions were similarly surveyed for all previously considered regions of interest (ROI0-ROI3), no different systematic response behaviour between the defined regions was actually found.

Overall, regions of interest are oscillating almost in phase (within about $\pm 30^\circ$) and exhibit a comparable amplitude ($\pm 15\%$) within which the fitted curves of the normalized intensity show smaller variations than the curves of its mean position.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figures}
\caption{Averaged normalized intensity of the region of interest ROI1 and ROI2 (top row) and the corresponding result of the sine curve fitting within all investigated ROI of the phase-averaged OH chemiluminescence and OH-PLIF intensity images for an operating condition (LCC) of $\lambda_{\text{eff}} = 2.0$, $T_{\text{mix}} = 650[K]$ and $m_{\text{air}} = 40[g/s]$.}
\end{figure}
By comparing the results derived from the OH chemiluminescence and OH-PLIF images, an astonishing outcome is observed. Whereas the sine describing the mean intensity obtained from the chemiluminescence and PLIF images oscillate in phase, the sine waves of the averaged flame front position have a distinct phasing of $=180^\circ$ between the two measurement techniques.

Apart from this, the light intensity due to the OH radical can be deemed to be an indicator for the heat release rate as well. For laminar and premixed flames, the heat release rate has its maximum a few tenths of millimetres before the concentration of the OH radicals starting to increase. Since the flame speed for laminar methane flames is quite low ($S_L < 0.6\,[\text{m/s}]$, see [56]), the resulting time lag between the approximated heat release rate and the OH intensity signal has been taken into account by evaluating the Rayleigh integral. Fortunately, the turbulent flow field of the EV burner increases the turbulence intensity and thus elevates the flame speed at least by two orders of magnitude. Due to the turbulent combustion, the time lag between the heat release rate and the OH intensity is negligible for an assessment of the Rayleigh integral with the intensity signal. (See Hermann [40] for a detailed analysis of stable turbulent premixed flames).

Accordingly, the phase angle of the coupling between the pressure oscillation and the OH intensity for the operating condition used to obtain Figure 5.36 results in a positive Rayleigh index, thus forcing the unstable, thermoacoustic tendency of this operating condition. An additional evidence of the positive Rayleigh index for this operating condition is similarly deducible from OH-chemiluminescence measured with the photomultiplier already used to characterise the distinct flame types in chapter 4. The graph of Figure 5.37 points at this issue.

![Figure 5.37](image-url)

**Figure 5.37.** Dynamic pressure signal (MIC1-2) of the combustion chamber and the plenum (MIC3) related to the filtered OH-chemiluminescence measured with a photomultiplier tube (PM) for an operating condition (LCC) of $\lambda_{afr} = 2.0$, $T_{\text{mix}} = 650\,[K]$ and $m_{\text{air}} = 40\,[\text{g/s}]$ similar to Figure 5.36.
A summary of the parameters, such as the amplitude $A_{fit}$, the offset $B_{fit}$ and the additional phase angle $\phi_{fit}$ of the sine wave function used to qualify the dynamic flame response of the investigated operating conditions (according to Figure 5.1), is shown in Figures 5.38, 5.39 and 5.40.

The different characteristics of the flame response for the short and the long combustion chamber are depicted in Figure 5.38. The dynamic flame motion, indicated by the averaged flame front position as well as the oscillating heat release derived from the chemiluminescence imaging technique, outstandingly coincides with the regions of the distinct flame types previously defined in chapter 4 by means of acoustic devices (sound pressure level SPL of MIC1-4), static pressure probes (pressure drop across the flame $\Delta p_{flame}$) and the mean and root mean square of the photomultiplier signal respectively ($PM_{mean}$ and $PM_{rms}$). The different flame types are indicated in Figure 5.38, in the graph showing the amplitude of the approximated sine function. Following the line of a constant mixture temperature $T_{mix} = 700[K]$ investigated in the long combustion chamber, the weak flame of flame type (IIa) ($\lambda_{afr} = 2.2$ and $\lambda_{afr} = 2.4$) shows a small amplitude of the averaged intensity oscillations as well as negligible fluctuations of the mean flame front position. In contrary, the unstable flame of flame type (IIb) has a major dynamical flame response for the mean intensity as well as for the mean flame front position. In the same way, the strongly burning flame (IIc) possesses reduced amplitudes of both investigated oscillations. Generally, this behaviour is consistent with all investigated mixture temperatures and in good agreement with the results presented in chapter 4.

In order for the Rayleigh index to be positive, the phase angle $\phi_{fit}$ has to lie within the range of $[0 - 180^\circ]$ and an angle of $90^\circ$ is required for an optimal forcing of the combustion. As can be seen from the phase angles of the normalized intensity, this fact is necessary, but seems to not to be sufficient for a thermoacoustically unstable condition. So, an operating conditions with a phase angle within the region of $[0 - 180^\circ]$ is not necessary unstable (e.g. flame type (IIa)), whereas an unstable flame definitely shows a phase angle slightly smaller than $90^\circ$! The chemical time scale, not shown but inherent to Figure 5.38 is mainly influenced by the air/fuel equivalence ratio and the mixture temperature (position of the flame stabilization, flame speed, etc.). Both parameters affect acoustic properties within the combustor as well (altered speed of sound and reflection coefficients). Additionally, the mixture temperature changes the hydrodynamic time scale by density effects (e.g. flow velocity). However, if these time scales come in agreement in an unfavourable way and clump into the acoustic feed back process, the instability is forced and an unstable flame results. For instance, this “clumping” is effective for the unstable operating condition of $\lambda_{afr} = 2.0$ and $T_{mix} = 650[K]$ (IIb) for the long combustion chamber. Since the primary difference between the long and
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Figure 5.38. Summarized results of the sinusoidal fit of phase-averaged OH chemiluminescence of the long and short combustion chamber (LCC/SCC) for a constant mass flow rate of $m_{\text{in}} = 40 \text{[g/s]}$.

Left: Parameters resulting of the averaged flame front position (ROI2).
Right: Parameters of the mean intensity within the region of interest (ROI2).
short combustion chamber is only the change of acoustic behaviour, the chemical and hydrodynamical time scales within the reaction zone are not affected by the altered combustor length. Indeed, the similar time scales do not yield the flame to be unstable due to the ineffective acoustic response. By changing the operating conditions of the SCC, another set of $\lambda_{afr}$, $T_{mix}$ and $m_{air}$ than for the LCC results in an effective hooking in of the acoustic property with the modified time scales.

Figure 5.39 is a further evidence of the prime importance of the involved time scales. The graphs are showing two distinct operating conditions, both having a constant mixture temperature and air/fuel equivalence ratio. Since $\lambda_{afr}$ and $T_{mix}$ have been fixed for these operating conditions, especially the hydrodynamic time scale is affected by the changing mass flow rate only. For unstable flame types $\lambda_{afr}$ and $T_{mix}$ shown in Figure 5.39, the hydrodynamic time scale affects the characteristic time of the feedback process responsible for the appearance of thermoacoustic instabilities. By considering the flame type $\lambda_{afr}$ of the LCC, for instance, the altered mixing rate at higher mass flow rates changes the characteristic time of the heat release fluctuations, which may result in a less effective coupling with acoustic properties of the combustion chamber; thus an increased mass flow rate stabilises this operating condition. However, an increasing mass flow rate results not strictly in a less effective coupling, as this example shows for the long combustion chamber. As can be seen from the behaviour of the SCC, an elevated mass flow rate shifts this time scales and induces an effective coupling to force thermoacoustic instabilities at higher flow rates (flame type $\lambda_{afr}$). It goes without saying, that the feedback cycle is established at higher frequencies for the SCC since the eigenfrequencies of the combustor are shifted to high values too.

Finally, Figure 5.40, points out differences of the measurement techniques (chemiluminescence and PLIF) for the different operating conditions. In general, the dynamic response of the flame investigated with two methods exhibits a similar behaviour for the amplitude as well as for the offset for all examined operating conditions. Whereas the offsets have almost identical values, the amplitudes deducted from the PLIF images are primarily smaller (ca. by a factor of 2). A single methodical difference could be explored by regarding the phase angle of the averaged flame front position of richer operating conditions. However, this distinct phasing of $\approx 180^\circ$ between the two measurement techniques has already been reported within this section.
Figure 5.39. Summarized results of the sinusoidal fit of phaseaveraged OH chemiluminescence images for the variation of the mass flow rate of the long and short combustion chamber (LCC/SCC). Left: Parameters resulting of the averaged flame front position (ROI2). Right: Parameters of the mean intensity within the region of interest (ROI2).
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Figure 5.40. Summarized results of the sinusoidal fit of phaseaveraged OH chemiluminescence and OH-PLIF images of the long combustion chamber (LCC) for a constant mass flow rate of $m_{\text{air}} = 40$ kg/s.

Left: Parameters resulting of the averaged flame front position (ROI2).
Right: Parameters of the mean intensity within the region of interest (ROI2).
6 Active Flame Stabilization

6.1. Air/fuel equivalence ratio fluctuations

Many recent studies conducted with lean premixed gas turbine burners pointed out that equivalence ratio fluctuations are a main cause of combustion instabilities ([54], [59], [64], [86]). If a perturbation of the air/fuel equivalence ratio is already present in the premixed section of the combustor, the rate of the heat release will be strongly influenced by the inhomogeneous mixture. However, a periodic oscillation of $\lambda_{afr}$ will excite a periodic heat release and the acoustic feedback process may influence and fortify the initial perturbation of $\lambda_{afr}$ in the section where the air and fuel-mixture generation occurs. The fluctuation of the mixture strength is then convected by the mean flow through the combustion chamber to the reaction zone.

6.1.1. Acetone LIF technique applied in the upstream section

To determine the role of air/fuel equivalence ratio fluctuations and possible variations of the power density of the flux as driving mechanisms of thermoacoustic instabilities, the upstream section of the combustion chamber has been investigated in more detail. A laser-induced-fluorescence technique was used to obtain the concentration of acetone in the upstream flow field located several diameters after the air/fuel mixture has been generated (See Figure 3.11, left). In advance, the fuel feed line has been seeded with acetone and a gaseous and homogenous composition of acetone and natural gas were preserved. Therefore, the concentration of acetone is directly proportional to the concentration of natural gas and a visible variation of the intensity due to fluorescence of the acetone molecule connotes also a variation of the concentration of natural gas.

Figure 6.1 shows a phase-averaged series of acetone LIF intensity images acquired in the upstream section of the combustion chamber. The first dominant mode of the pressure wave present in the upstream section was used to trigger the laser system and the image acquisition. Similar results were achieved by triggering the system with the first dominant mode present in the downstream section. The intensity of the images was shot-to-shot corrected with respect to the fluctuating energy of the laser beam. The
Air/fuel equivalence ratio fluctuations

Figure 6.1. Images of acetone LIF intensity (normalized over the phase) for the long combustion chamber (LCC) measured in the upstream section for an operating condition of $\lambda_{af} = 2.0$, $T_{\text{mix}} = 700[K]$ and $m_{\text{air}} = 40[g/s]$.

Laser sheet inhomogeneity was not adjusted for these images to prevent additional noise introduced by the correction procedure. Apart from the stripes generated by the laser sheet, coherent structures of the flow, such as wakes or even inhomogeneous intensity regions produced by a changing concentration of acetone could not be observed in the upstream section of the combustion system. The Figure 6.2 (left) is obtained by integrating the intensity within a phase-averaged image, normalizing this value with the maximal intensity in the series and finally, by plotting the resulting intensity for different phase angles. The figure on the right hand side shows the minimal, maximal and averaged intensity of every image. As a major consequence from this figure, it can be stated that a periodic fluctuation of the intensity signal is not noticeable and therefore no air/fuel equivalence ratio fluctuations should be present in the upstream section. This behaviour was checked for several operating conditions including stable and unstable flames. In addition, to give further proof of this statement and to assess the validity of results, a second measurement technique has been applied to track possible equivalence ratio perturbations in the upstream part of the combustion chamber.

Figure 6.2. Left: Averaged intensity of a region of interest for the images showing the acetone LIF intensity (normalized over the phase) for the long combustion chamber (LCC) measured in the upstream section for $\lambda_{af} = 1.8$, $T_{\text{mix}} = 650[K]$ and $m_{\text{air}} = 40[g/s]$.

Right: Minimum, maximum and mean intensity of every image within a period of the dominant frequency of the unstable operating condition.
6.1.2. Methane absorption technique in the up- and downstream section

The methane absorption technique, as an established method to measure methane concentrations and as a simple line of sight method, has been used to systematically investigate air/fuel equivalence ratio fluctuations in the parameter space defined by the overall air/fuel equivalence ratio $\lambda_{afr}$ for a range of 1.70 up to the lean extinction limit and for mixture temperatures within the range of $T_{mix} = [500 - 750]$K. Measurements comprised two different mass flow rates ($m_{air} = 36$[g/s] and $m_{air} = 40$[g/s]) and were only conducted for the configuration with the long combustion chamber. The axial position of the laser beam crossing the upstream section was selected similar to the zero position of the acetone LIF intensity images shown in the previous subsection (Figure 6.1). Additional measurements were performed at an axial distance of $\pm40$[mm] to rule out the possibility of unfortunately measuring at the location of a pressure node resulting in a wrongly detected constant mixture fraction (for the implausible case, where $\lambda_{afr}$-fluctuations are not simply convected). Furthermore, the air/fuel equivalence ratio fluctuations were additionally measured in the downstream section in a plane close at the burner exit at a distance smaller than $2$[mm].

In order to summarize these results, the root mean squared value of the non-absorbed signal obtained with a photovoltaic infrared detector ($PID_{rms, normalized}$) has been calculated and normalized with the maximal averaged value measured in the upstream and downstream section for a constant mass flow rate. In fact, the previous statement is corroborated. It can be seen from the right side of Figure 6.3, that no detectable air/fuel equivalence ratio fluctuations were present in the upstream section, even for colder (and richer) operating conditions, where major pressure amplitudes dominate the combustion process. Unlike to the upstream section, the concentration of methane in the downstream section started fluctuating for operating conditions corresponding to

![Figure 6.3](image-url)
strong thermoacoustic instabilities where the flame structure with their slow transition was identified (See Chapter 4.2). The slow transition was previously characterized with large pressure oscillations (1st longitudinal mode) and extensive movement of the flame front location. Fluctuating absorption values cannot be the result of a dominant oscillation of the global air/fuel equivalence ratio (such oscillations would be noticed in the upstream section). Oscillations are more the consequence of the movement of the flame front location. The reaction zone and its burnt and unburnt sections are crossing the line of sight of the laser beam and produce periodic variations of the CH4 absorption. Figure 6.4 supports this argument by showing two different pressure and two absorption spectra, each belonging to the upstream and downstream part of the combustor and in addition, the spectrum of the OH filtered photomultiplier signal is given to illustrate the flame movement.

Figure 6.4. From upper left to lower right: SPL of MIC2, SPL of MIC4, Signal of the photovoltaic detector (PID) measured in the upstream section and at the burner exit and the PM signal. All spectra are functions of the Strouhal numbers St for constant operating condition at $\lambda_{air} = 1.8$, $T_{mix} = 600[K]$ and $\dot{m}_{air} = 40[l/s]$. 
6.2. Characteristics of a secondary fuel injection system

The fundamental idea to stabilize oscillations of the heat release rates by a secondary fuel injection is heavily dependent on the controller effectiveness of the deployed actuator. Additional fuel line dynamics and delays by the mixing process (including a convective time lag) of the gas with the air as well as the attenuation of the modulated gas flow significantly reduce control authority. Therefore, the basic performance of the actuator used to modulate the secondary fuel injection and its influence on the reaction zone is consequentially discussed within this section.

6.2.1. Transfer functions of the actuator

The mass flow rate of a secondary fuel injection was intended to be kept, if at all, as small as practicable, but in any event not to fall below a certain limit to prevent a weakening of control authority. At a standard operating condition of an air mass flow rate of $m_{\text{air}} = 40\text{[g/s]}$ and an air/fuel equivalence ratio of $\lambda_{\text{afr}} = 2.0\text{[-]}$, the total gas mass flow has a resulting nominal value of $m_{\text{gas}} = 1.2\text{[g/s]}$. 10% of this main gas flow was chosen as a lower design point for the secondary fuel injection rate, where $m_{\text{gas}} = 1\text{[g/s]}$ was set as mass flow rate at full stroke of the valve. Figure 6.5 shows the characteristic curve of the GMM valve for different values of the forcing offset $O_{\text{forcing}}$ within the range of $[0 - 1]\text{[V]}$ with a norm pressure across the valve of $\Delta p_{\text{valer, norm}} = 2.4\text{[bar]}$. The non-linear behaviour in the upper region ($[0.7 - 1]\text{[V]}$) between the forcing offset and the mass flow rate used by the secondary fuel injection is due to a saturation effect of the flow metering system for a higher fuel gas throughput. Within the range of a forcing offset $O_{\text{forcing}} = [0.25 - 0.75]\text{[V]}$ the mass flow rate has a strictly linear characteristic. Modulating the flow beyond this border by driving the valve forcing offset $O_{\text{forcing}}$ and/or amplitude $A_{\text{forcing}}$ to higher or lower levels, results in an overrated mass flow rate.

![Figure 6.5](image_url)
Since every actuator has a characteristic frequency response and an individual time delay with respect to the input set point signal, the dynamic behaviour of the GMM valve has been measured. For the dynamic characterization of the valve, a HP36650 dynamic signal analyser sent a swept sine signal to the input and monitored the output signal of the examined system to derive a transfer function. The signal analyser, the measurement technique, as well as further results are accurately described and reported by the investigations carried out by Niederberger et al. ([68], [69]). Figure 6.6 shows the transfer function derived between the commanded voltage and the rod position of the valve measured with a displacement sensor. As can be seen from the right side of the graph, the resonant frequency of the mechanical response is outstanding compared

Figure 6.6. Two distinct transfer function of the GMM valve identified between the excitation signal and the displacement sensor for a fixed offset \(O_{\text{forcing}} = 0.3[V]\) and amplitude of \(A_{\text{forcing}} = 0.3[V]\) (Right figure: \(O_{\text{forcing}} = 0.6[V], A_{\text{forcing}} = 0.15[V]\)).

Figure 6.7. The transfer function of the GMM valve identified between the excitation signal and the signal of the photovoltaic infrared sensor for a fixed offset \(O_{\text{forcing}} = 0.3[V]\) and amplitude of \(A_{\text{forcing}} = 0.3[V]\). The absorption of methane is measured in the free jet close (\(3[mm]\)) after the exit of the tube (left) and at a distance of \(30[mm]\) (right).
to direct drive servo valves with a typical frequency lower than $f_{rel} = 500 \text{[Hz]}$ ([4],[18],[38],[111]). If the valve is operated in the linear regime of Figure 6.5, the resonant frequency lies clearly above $1500 \text{[Hz]}$. Even for operating conditions where a part of the modulation falls in the non-linear regime (Figure 6.6), the transfer function has a pleasant characteristic below $1000 \text{[Hz]}$.

However, a formidable mechanical transfer function does not guarantee for a sufficient modulation of the flow and especially attention should be paid to the fact, that the flow is modulated in the gas phase. As a consequence, the impact of the rod to the gas and the ability to modulate the flow are less effective resulting in a reduced control authority. For the dynamic characterization of the flow through the valve, the already presented methane absorption technique has been used (see “Methane absorption measurement technique” on page 39). The methane flow modulated by the valve has been investigated after a $50 \text{[mm]}$ long air duct at two distinct levels of the free jet ($3 \text{[mm]}$ and $30 \text{[mm]}$). Figure 6.7 shows the transfer function between the valve command and the signal of the photovoltaic infrared sensor measuring the different absorption rates due to the modulated flow rate. As can be seen from the transfer function, obtained close at the exit of the tube, an excellent characteristic is achieved up to $300 \text{[Hz]}$, whereon a sharp decline a steep rise is followed at a frequency identical to the resonant frequency of the mechanical system. The decline in the magnitude plot is smoothed and less alleviated ($-3 \text{[dB]}$ at $1000 \text{[Hz]}$) as soon as the excitation signal is strictly maintained in the well behaved linear regime of the characteristic curve shown in Figure 6.5.

6.2.2. Acetone LIF technique applied at the exit of the actuator

In order to reassess an initial concern that the valve is not able to modulate the gas flow rate for forcing frequencies higher than $250 \text{[Hz]}$, a second measurement method was applied to investigated the frequency behaviour of the free jet. Therefore, the gas flow was seeded with acetone and the 2D imaging technique already presented in Chapter 3.4.3 (Acetone-Laser Induced Fluorescence) was used at the exit of the valve. Several operating points composed of a sinusoidal input with a specific forcing frequency ($\omega_{\text{forcing}} = [100, 200, 300, 400, 500, 750, 1000, 1500, 2000, 3000, 4000] \text{[Hz]}$), two different forcing amplitudes ($A_{\text{forcing}} = [0.15, 0.3] \text{[V]}$) and three mean command values for the valve ($O_{\text{forcing}} = [0.3, 0.4, 0.6] \text{[V]}$) were investigated. For the 66 resulting operating conditions, the free jet was examined at eight phase angles and the average intensity of the acquired images was calculated for every individual phase.

An exemplary result for a forcing frequency of $300 \text{[Hz]}$ is depicted in Figure 6.8 (See the appendix for the other frequencies at $100 \text{[Hz]}$, $200 \text{[Hz]}$ and $500 \text{[Hz]}$). The movement of the injection front is highly visible as well as the intensity variations. The phase
delay due to the transport lag in the tube between the methane absorption measurement and the acetone-LIF technique matches quite well. The new injection front is introduced in the ambient air at phase angle of $\phi_{\text{trig}} = 135^\circ$ which corresponds to the phase delay estimated within previous transfer function measurements.

In summary, it can be ascertained that a movement of the injection front and/or a variation of the intensity is limited and not visible for forcing frequencies higher than 750 [Hz]. The peak in amplitude of the resonant frequency at 2000 [Hz] could not be verified.

Larger amplitudes as well as higher offset values increase perceived intensity levels of the images and thus influence the control authority favorably. Similar results are shown in more details in section 6.2.5.

Figure 6.8. Phase-averaged LIF intensity (normalized over the phase) images of the free jet acquired at the exit of the valve. The valve was operated at a sinusoidal forcing frequency of $f_{\text{forcing}} = 300$ [Hz] and at an amplitude of $A_{\text{forcing}} = 0.3$ [V]. The mean command voltage was set to $O_{\text{forcing}} = 0.3$ [V] and the gas flow was seeded with acetone prior to the valve.
6.2.3. Transfer functions of the actuation system

A crucial factor for success of the secondary fuel injection strategy is the ability to transport the modulated gas flow as expeditiously as possible to the reaction zone, to prevent dissipation losses of the modulation amplitude. On the other hand, a complete mixing of fuel and air shortly after the injection prevents an increase of \( NO_x \) levels and averts the worst case scenario, the production of soot emissions. However, a compromise must be found and realized since a better mixing process of fuel and air is strongly associated with a higher attenuation of the injection profile resulting in a reduced control authority.

For this type of combustor, the effects of injection parameters such as the position of the injection, the injection profile and the duration as well as the influence of the mass flow rate are investigated within a basic study by I. Kaiktsis et al. [45] using computational fluid dynamics (STAR-CD, cold flow RANS simulation) for the geometry of the burner employed here. On the basis of these calculations, a secondary fuel injection strategy located upstream of the combustor and characteristic mixing times in the order of milliseconds implicated therein, could not be effective for a control strategy due to a vastly convective time lag. Therefore, a solution of injecting in or downstream of the burner seems to be the only feasible way to obtain a transfer function qualifying for an effective feedback control of thermoacoustic instabilities.

The two half-cones of the EV burner include two fuel distribution tubes close at the air inlet slot. At this diametrically opposed air inlet slots, the fuel is injected into the air stream along equidistant gas injection ports (orifices) and a complete mixing of fuel and air is obtained within a few millimetres after the injection. This injection scheme has been rebuilt with two smooth bore pipes, each for one side of the half cone, to remove dead spaces, obstacles and edges in order to avoid unwanted acoustic reflections and not to affect the dynamics of the transport of the modulation negatively. The arrangement of the injection ports has been retained the same for an analogue repro-

![Figure 6.9](image-url) **Figure 6.9.** Different gas paths for the secondary fuel injection strategy including the original gas injection system (A) of the EV burners where all orifices are used and with a two orifices configuration (B).
Characteristics of a secondary fuel injection system

duction of the EV burner (Figure 6.9, A). During this work, it was necessary to modify the arrangement of the injection ports. Thus, a second burner was equipped with two smooth bore pipes on every side of the burner similar to the original configuration of Figure 6.9 (A). The significant difference between the two configurations consists of the distinct arrangement of the orifices. The advanced strategy (Figure 6.9, B) includes an EV5 combustor with only two ports for the secondary fuel injection located close at the dump plane.

Similar to the previous transfer function measurement, the methane absorption technique was used to characterize the dynamic behaviour of the flow through the valve and the piping system of the distinct fuel injection strategy. The laser beam, differently absorbed by a variable concentrations of methane and detected by an infrared sensor, crossed the exit of the EV5 burner in respect of the dump plane at a separation distance smaller than 2[mm]. The test rig was rinsed with an air mass flow rate of $n_{\text{air}} = 36[g/s]$ (at a mixture temperature of $T_{\text{mix}} = 700[K]$) to prevent a continuous filling of the combustion chamber with methane and a steady growing concentration. Figure 6.10 shows the resulting transfer functions for both, the EV burner with the multi-hole and the two-hole configuration. It can be concluded from the figures, that the response for low forcing frequencies is more significant ($+20[dB]$) by using the two-hole configuration. For higher frequencies, both transfer functions exhibit large fluctuations (increase of the measurement uncertainty) in the magnitude as well as in the phase plot. However, the methane absorption method, as a line of sight technique, seems not to be an adequate means of measuring non uniform concentrations of methane at the combustor exit. Further investigations related to the mixing characteristics and the possibility to excite the system with the GMM valve is given in next subsection.

![Figure 6.10](image.png)

**Figure 6.10.** Transfer function of the GMM valve operated at a fixed command offset $O_{\text{forcing}} = 0.3[V]$ and amplitude of $A_{\text{forcing}} = 0.3[V]$. The transfer function is identified between the excitation signal and the signal of the photovoltaic infrared sensor mounted close at the burner exit for the two-hole (left) and the multi-hole configuration (right).
6.2.4. Acetone LIF technique applied at the exit of the EV5 combustor

Figures 6.11 and 6.12 show acetone LIF intensity images acquired downstream of the EV burner at the dump plane for the two different injection schemes (multi-hole / two-hole configuration). The main air flow rate was adjusted to $m_{air} = 36[g/s]$ and to a temperature of $T_{mix} = 500[K]$. The main gas flow rate was set to zero while on the other hand, the gas flow through the valve was seeded with acetone and modulated with the GMM valve ($m_{valve} = 0.57[g/s]$).

All phase-averaged images, the multi-hole injection as well as the two-hole injection configuration have an asymmetric intensity distribution relative to the center axis ($x = 0$) of the burner. This can be attributed to the fact that, for the left side of the image (of the flow field after the burner), the acetone-gas mixture released from the left multi-hole piping is transported to the laser sheet, whereas for the right side of the

![Phase-averaged acetone LIF intensity images](image)

**Figure 6.11.** Phase-averaged acetone LIF intensity (normalized over the phase) images acquired at the exit of combustor downstream of the dump plane for a rinsing flow of $m_{air} = 36[g/s]$ and a temperature of $T_{mix} = 500[K]$. The valve itself was operated with the *multi-hole injection* configuration at a sinusoidal forcing frequency of $f_{forcing} = 300[Hz]$, with an amplitude of $A_{forcing} = 0.3[V]$ and a mean command voltage of $O_{forcing} = 0.61[V]$ ($m_{valve} = 0.57[g/s]$).
images the mixture does not cross the laser sheet. At this area, a stream of “fresh” air enters the chamber resulting in a lower intensity due to a minor concentration of acetone. However, a modulation of the recorded intensity and/or a displacement of an intensity region is not observable. The numerous axial positions of the injection ports along the EV burner result in a smeared forcing amplitude, especially for higher forcing frequencies. Tests in the reactive flow underlay this statement by showing a weak down to a missing control authority depending on the gas flow rate used by the secondary fuel injection.

In contrast to the multihole configuration, the two-hole configuration shows an improved characteristic (Figure 6.12). Although, the asymmetric intensity distribution is invariably present, the intensity modulation within a forcing period due to the GMM valve is clearly visible. The flow stream of the acetone-gas mixture, pouring out of the the left and right injection orifice, is visualized (indicated with black arrows) as

![Figure 6.12](image)

Figure 6.12. Phase-averaged acetone LIF intensity (normalized over the phase) images acquired at the exit of combustor downstream of the dump plane for a rinsing flow of $m_{\text{flow}} = 36$[g/s] and a temperature of $T_{\text{inlet}} = 500$[K]. The valve itself was operated with the two-hole injection configuration at a sinusoidal forcing frequency of $f_{\text{forcing}} = 300$[Hz], with an amplitude of $A_{\text{forcing}} = 0.3$[V] and a mean command voltage of $O_{\text{forcing}} = 0.6$[V] ($m_{\text{valve}} = 0.57$[g/s]).
well as the increase and decrease over a period of the concentration of acetone inside of the inner recirculation zone (see white arrows). The main part of the air flow is entering the downstream section of the combustion chamber through the slots of the EV combustor without any mixing with acetone and afterwards further downstream, the entire flow is mixed up. By considering iso-lines of a constant intensity within the region of lower concentrations of acetone (blue regions), a bimodal distribution is obtained, similar to the typical curvatures of the flame front (See Chapter 5).

6.2.5. Influence of the operating conditions (steady state)

The mixture temperature $T_{\text{mix}}$, the mass flow rate of the main air flow $m_{\text{air}}$ and the gas flow through the GMM valve $m_{\text{valve}}$ are steady state parameters that influence the properties of a secondary fuel injection for non-reactive operating conditions. Hence, a careful investigation was carried out for the two-hole injection configuration to characterize the influence of the above mentioned parameters on the mixing process in the flow field close to the two injection ports. The secondary mass flow rate was seeded with acetone and the resulting concentrations visualized downstream of the EV burner (acetone LIF technique). The intensity of the images was normalized with the identical factor used to scale the images of Figure 6.12. The most important results of the variation of the main mass flow rate $m_{\text{air}}$, of the mixture temperature $T_{\text{mix}}$ and the command offset are set out in Figures 6.14 and 6.15.

![Image](image-url)

Figure 6.14. Untriggered intensity images (acetone LIF) showing the region of interest ROI2 for several operating conditions (all including a secondary fuel injection with the two-hole design):

- Top: $m_{\text{air}} = [30, 34, 38, 42, 46] \text{[g/s]}$, $T_{\text{mix}} = 500\text{[K]}$, $m_{\text{valve}} = 0.57\text{[g/s]}$.
- Middle: $T_{\text{mix}} = [500, 550, 600, 650, 700, 750]\text{[K]}$, $m_{\text{air}} = 46\text{[g/s]}$, $m_{\text{valve}} = 0.57\text{[g/s]}$.
- Bottom: $\Delta_{\text{forcing}} = [0.3, 0.4, 0.5, 0.6, 0.7, 0.8] \text{[V]}$, $m_{\text{air}} = 46\text{[g/s]}$, $T_{\text{mix}} = 500\text{[K]}$. 

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The variation of the offset results in a mean flow rate $\dot{m}_{\text{valve}}$ through the GMM valve of $\dot{m}_{\text{valve}} = [0.05, 0.26, 0.43, 0.57, 0.71, 0.79] \text{[g/s]}$. The region of interest ROI1 (this region of interest is identical to the area of an image depicted in Figure 6.12), ROI2 and ROI3 exhibit similar trends and only an intensity offset between the different regions is observable. Higher air mass flow rates of the main flow through the combustor as well as higher temperatures of the air raise the flow velocity and transport rates increase respectively. The concentration of acetone is thinned out in the region of interest and a lower intensity value is ensued. On the one hand, the higher velocity of the flow acts positively by reducing the convective time lag and otherwise negatively to the control authority. The increased velocity gradient between the secondary gas flow and the main flow through the slots of the EV combustor increase the mixing rate (turbulence effect) and therefore attenuate the forcing amplitude by smoothing concentration gradients.

![Graphs showing the variation of intensity values in different regions](image)

**Figure 6.15.** Averaged and normalized intensity values for different region of interests resulting from images acquired with the acetone LIF technique at the following operating conditions (all including a secondary fuel injection with the two-hole design):

- **Top:** $\dot{m}_{\text{air}} = [30, 34, 38, 42, 46] \text{[g/s]}, \; T_{\text{mix}} = 500[\text{K}], \; \dot{m}_{\text{valve}} = 0.57 \text{[g/s]}$.
- **Middle:** $\dot{m}_{\text{air}} = [1500, 550, 600, 650, 700, 750] \text{[g/s]}, \; m_{\text{air}} = 46 \text{[g/s]}, \; \dot{m}_{\text{valve}} = 0.57 \text{[g/s]}$.
- **Bottom:** $\dot{O}_{\text{forcing}} = [0.3, 0.4, 0.5, 0.6, 0.7, 0.8] \text{[V]}, \; \dot{m}_{\text{air}} = 46 \text{[g/s]}, \; T_{\text{mix}} = 500[\text{K}]$.
6.3. Flame response of a secondary fuel injection

In face of all previous investigations, reactive experiments with a secondary fuel injection to force the flame are still the most realistic cases to check the performance and to evaluate any unwanted influence of the injection strategy on the flame. As result of chapters 6.2.3 and 6.2.4, the secondary fuel injection system with the two ports close to the dump plane of the EV burner (two-hole configuration) shows the most promising outcomes for a successful implementation of an active instability control of the reaction zone. Therefore, the current discussion and the following subchapters are restricted to the two-hole control strategy to simplify matters. For the further steps, the frequency as well as the amplitude and the actual phase of the forcing signal are not consciously synchronized with the natural frequencies appearing in the combustion system. The ability to influence, or even destroy, the natural characteristics of the flame structure shed therefore light on the magnitude of the control authority.

The measurements of the OH chemiluminescence intensity obtained with the long combustion chamber underlay a strong influence of the gas mass flow of the secondary fuel injection resulting in a modified flame characteristic. Figure 6.16 shows not synchronized intensity images for two different operating conditions at a constant mixture temperature of $T_{\text{mix}} = 700[K]$, at a main air-mass flow rate of $m_{\text{air}} = 36[g/s]$ and in particular, the image on the right of Figure 6.16 includes the usage of the secondary fuel injection system. The gas flow used by the secondary injection is subtracted by a control algorithm from the main fuel feed line, whereby the overall air/fuel equivalence ratio is kept constant at a level of $\lambda_{\text{airf}} = 2.2$ for both cases. A tendency to cause asymmetric flame shapes due to the secondary injection is obvious and strongly related to increasing (secondary) gas mass flow rates and the already present flame type.

![Figure 6.16](image)

**Figure 6.16.** Time-averaged images of the OH chemiluminescence intensity for a constant mixture temperature $T_{\text{mix}} = 700[K]$, a mass flow rate of $m_{\text{air}} = 36[g/s]$ and an air/fuel equivalence ratio of $\lambda_{\text{airf}} = 2.2$. Using the secondary fuel injection at forcing frequency of $f_{\text{forcing}} = 200[Hz]$, an offset of $\phi_{\text{forcing}} = 0.4[\pi]$ ($m_{\text{inlet}} = 0.26[g/s]$) with $A_{\text{forcing}} = 0.3[\pi]$. 139
6.3.1. Transfer functions of the actuation system including combustion

From a control engineering point of view, a complete characterisation of the secondary fuel injection system is obtained by measuring the transfer functions between the forcing command voltage to drive the GMM valve and the resulting pressure fluctuations (MIC2) produced by the forced reaction zone. In respect thereof, several operating conditions of the combustor as well as different forcing procedures (command offsets and amplitudes) using the secondary fuel were investigated.

Remarkable is the fact that the flame structure, depending on the overall operating parameters of the combustor such as $T_{\text{mix}}$, $\dot{m}_{\text{air}}$ and $\lambda_{\text{air}}$, strongly influence the dynamic performance of the secondary fuel injection (See Figure 6.17). The convective time lag of a fluid particle moving from the injection ports to the reaction zone is significantly larger for leaner operating conditions (flame type $\text{Ib}$), such as those for

![Figure 6.17](image)

**Figure 6.17.** Transfer functions of the GMM valve (two-hole configuration) operated at a fixed command offset $\Delta V_{\text{control}} = 0.3 \text{[V]}$ ($\dot{m}_{\text{air}} = 0.26 \text{[g/s]}$) and amplitude ($\Delta V_{\text{command}} = 0.3 \text{[V]}$). The transfer function is identified between the excitation signal and the pressure response of a microphone (MIC2) for an operating condition at $T_{\text{air}} = 700 \text{[K]}$, $\dot{m}_{\text{air}} = 36 \text{[g/s]}$, $\lambda_{\text{air}} = 2.4$ (upper left), $\lambda_{\text{air}} = 2.2$ (upper right) and $\lambda_{\text{air}} = 2.0$. 
richer flames (\( \text{(IIa)} \)). Hence, the additional phase shift is further reduced by forcing even richer flames structures (\( \text{(IIb)} \) and \( \text{(IIa)} \)). In contrary, magnitude plots associate a stronger control authority in the lower frequency range with the flame type of \( \text{(IIa)} \) (and \( \text{(IIb)} \)). These flame structures exhibit already a “natural” substantial movement of the flame front (along with an increase of the dynamic pressure levels) traced back to the thermoacoustic instability. However, these flame types are more susceptible to a forcing by the secondary fuel injection in almost the same manner. On the other hand, richer operating conditions generate a higher pressure feedback of the forcing amplitude in the higher frequency range than flames operated at leaner air/fuel equivalence ratios (compare resonant peak at \( 700 \text{[Hz]} \)).

6.3.2. Phaseaveraged OH chemiluminescence images of the forced flame

The transfer functions derived for the distinct flame structures shown in Figure 6.17 have a common decay of the forcing response of the amplitude (see magnitude plot) starting approximately at a frequency of \( 200 \text{[Hz]} \). With the help of the phaselocking imaging technique (OH chemiluminescence), the falling response characteristic of the secondary fuel injection system (two-hole configuration) for the frequency range of \( f_{\text{forcing}} = [100 - 500] \text{[Hz]} \) has been investigated for corresponding operating conditions with different mixture strengths.

An example of the phaseaveraged intensity images over an entire forcing phase is shown in Figure 6.18. Intensity values rise and fall with respect to the forcing signal that commands the GMM valve. Similar to the method presented in chapter 5, the position and the mean value of the normalized intensity of every image acquired at a specific phasing

![Phaseaveraged OH chemiluminescence images](image)

**Figure 6.18.** Phase-averaged images of the OH chemiluminescence intensity (normalized over the phase) for a constant mixture temperature \( T_{\text{mix}} = 700 \text{[K]} \), \( \dot{m}_{\text{air}} = 36 \text{[g/s]} \) and \( \lambda_{\text{air}} = 2.2 \) using the secondary fuel injection with the two-hole design at forcing frequency of \( f_{\text{forcing}} = 200 \text{[Hz]} \), a command offset of \( \phi_{\text{forcing}} = 0.4 \text{[\emph{V}]} \) \( (\omega_{\text{forcing}} = 0.26 \text{[g/s]} \) and with a forcing amplitude of \( A_{\text{forcing}} = 0.3 \text{[\emph{V}]} \).
Figure 6.19. Amplitude, phase and offset of the sinusoidal fit of the mean flame front position (left) and intensity (right) of the images obtained with the OH chemiluminescence technique for a constant operating condition ($T_{m,ix} = 700\,[K]$, $m_{mix} = 36\,[g/s]$, $\gamma_{eff} = 2.2$). The range of the forcing frequency of the secondary fuel injection (two-hole configuration) was varied between $f_{forcing} = 100\,[Hz]$ and $500\,[Hz]$ with a constant offset of $O_{forcing} = 0.4\,[V]$ ($m_{value} = 0.26\,[g/s]$) and a forcing amplitude of $A_{forcing} = 0.3\,[V]$. 

Flame response of a secondary fuel injection
Flame response to secondary fuel injection

($\phi_{\text{trig}}$) are calculated. Moreover, a sinusoidal least square fit is applied for every derived value (position and intensity), each one coinciding with a specific phase angle. The parameters deliverable from the fitting process are described by a sine wave in a universal form, whereby the angular frequency $\omega$ is set equal to $2\pi f_{\text{forcing}}$:

$$f(t) = A_{\text{sfit}} \cdot \sin(\omega t + \phi_{\text{sfit}}) + B_{\text{sfit}}, \text{ where } -\pi \leq \phi_{\text{sfit}} < \pi$$

(6.1)

Results achieved in the right region of interest (ROI3) are summarized in Figure 6.19 for the frequency range of $f_{\text{forcing}} = [100 - 500]\text{[Hz]}$. First, there is a gradual downward trend for the amplitude $A_{\text{sfit}}$ (modulations of the position and the mean intensity value) visible for higher forcing frequencies. The amplitude of the position of the mean intensity is stronger for leaner operating conditions and the $A_{\text{sfit}}$-values for the forced intensity fluctuations are located at the highest level for an air/fuel equivalence ratio $\lambda_{\text{af}} = 2.2$. This fact supports previous outcomes according to which operating conditions already being susceptible for thermoacoustic instabilities (slow transition) are easier to actuate by a secondary fuel injection (flame position and intensity). Flame structures appearing at leaner operating conditions ($\Theta$) respond strongly with a movement of the position of the intensity than with a variation of the intensity levels whereas the flame type $\Delta$ (richer $\lambda_{\text{af}}$) tends to act opposite (intensity variation instead of position).

And secondly, the behaviour of the phasing $\phi_{\text{sfit}}$ has been previously reported in chapter 6.3.1 in the same manner: flames types with a flame front located further downstream (correctly ordered from close to far: $\llbracket\Delta, \Theta, \llbracket\Gamma\rrbracket, \llbracket\Theta\rrbracket$) exhibit a steady rise in the convective time lag resulting in an additional phase delay. Finally, the forcing frequency of the secondary fuel injection does not seem to exert any influence on the offset value of the sine $B_{\text{sfit}}$ (position and intensity).

A variation of the forcing amplitude $A_{\text{forcing}} = [0 - 0.4]\text{[V]}$ of the secondary fuel injection system was carried out in addition to the above presented variation of the frequency. The command offset was chosen at a constant level of $O_{\text{forcing}} = 0.4\text{[V]}$ and the modulation was forced with a frequency of $f_{\text{forcing}} = 200\text{[Hz]}$. Similar to the previous investigation, the three considered operating conditions were tested and set to $T_{\text{mix}} = 700\text{[K]}$, $m_{\text{air}} = 36\text{[g/s]}$ and $\lambda_{\text{af}} = [2.0, 2.2, 2.4]$.

The above named findings are supported with these experiments without any exception and it was additionally noticed, that increased forcing amplitude causes a rise up to severer fluctuations (increasing $A_{\text{sfit}}$) of the mean intensity as well as to stronger modulations of the position of the mean intensity. Therefore, the decay of the modulation for higher frequencies could be compensated by increasing the command amplitude.
6.3.3. Influence of the operating conditions (steady state)

Previous outcomes of this work showed the influence of the gas flow rate used by the secondary gas injection system on the fuel concentration close to the injection ports even without modulating the secondary flow rate (see acetone LIF images of section 6.2.5). The different distribution of the fuel concentration will affect the reaction zone and yield a slightly modified position where the flame will be located in the flow field of the EV burner. For this reason, reactive tests with several constant command voltages were carried out to obtain a steady flow rate through the secondary injection system (two-hole configuration) and to denominate the influence on the flame characteristic with the help of the OH chemiluminescence imaging technique.

An example of this is given in Figure 6.20 for the position of the flame front and the center of gravity within different OH intensity images. The secondary fuel injection rate, with no forcing frequency and amplitude superposed on the mean flow, was varied from \( m_{\text{value}} = 0 \) [g/s] up to \( m_{\text{value}} = 0.57 \) [g/s] by changing the command voltage within the range of \( O_{\text{forcing}} = [0 - 0.6] \) [V]. The global operating condition was set to \( T_{\text{mix}} = 700 [K] \), \( \dot{m}_{\text{air}} = 36 [g/s] \), \( \lambda_{\text{afr}} = 2.4 [-] \) and serves in this place as a case point. The baseline condition (1), without any flow through the GMM valve, exhibits a flame structure belonging to flame type \( \text{(Ib).} \) However, the position of the flame front and, in the same manner the center of gravity are drawn closer to the combustor by an increasing fuel concentration introduced by the two injection ports. Although all of the global

![Figure 6.20](image_url)
parameters such as the air/fuel equivalence ratio were kept constant and just the ratio of the natural gas conducted through the secondary fuel injection system and the main gas feed line was altered, the flame structure is changed dramatically. The V-shape of the flame at the baseline condition is continuously opened out and transformed to a bimodal flame front (M-shape) while in parallel, the flame front is moved closer to the burner exit (the movement of the CoG shows the similar behaviour). In chapter 5, a strong dependency of the position and the shape of the flame front caused by an alteration of the air/fuel equivalence ratio, the mixture temperature as well as of the air mass flow rate were demonstrated. To this conclusion, the influence of an increasing, but steady rate of a secondary injection has to be added.

If the positions of the flame front resulting by an increasing rate of the secondary injection are compared to the corresponding positions resulting from the specific air/fuel equivalence ratios (by holding the mixture temperature and air mass flow rate constant), the movement of the flame front due to the secondary fuel injection can be quantified in terms of \( \Delta \lambda \). For the current case, a variation of the flame position by the secondary fuel injection rate of \( \frac{m_{value}}{m_{gas, tot}} = 0 - 50\% \) would represent an air/fuel equivalence ratio rise of \( \Delta \lambda = 0.4 \) (starting at the same baseline condition) to produce the same location of the flame front. Moreover, this implies an alteration of which flame structures starting at flame type [\( @ \)] through [\( @ \)], to [\( @ \)] and ending at [\( @ \)], which includes the passing through the fast transition line and the slow transition regime as well (See Figure 4.2 for flame structures). However, the degree of the movement for the flame front by the secondary fuel rate is strongly related to the settings of the global operating condition such as \( T_{mix}, m_{air}, \lambda_{afr} \) and within which type of flame the baseline condition without any secondary fuel flow lies.

An interlocking question of the flame front movement is the impact of this change on the sound pressure levels (SPL of MIC2) and on the emission level of the combustor (such as \( NO_x \)). As can be seen from Figure 6.21, operating conditions related to the flame types [\( @ \)] and [\( @ \)] (combustor fired at a richer air/fuel equivalence ratio \( \lambda_{afr} = 2.0 \) and \( \lambda_{afr} = 2.2 \)) possess a decreasing sound pressure level SPL for a raising secondary fuel injection rate. On the contrary, the lean operating condition (\( \lambda_{afr} = 2.4 \)) of flame type [\( @ \)] starts already at a silent precondition and a secondary fuel injection rate increases the sound pressure level at the same time. It has to be pointed out, that the sound pressure level is increased monotonically even though the flame front passes front positions, if operated at baseline conditions, were corresponding to the fast and the slow transition. For the minimal achieved sound level in the combustor, an asymptotic characteristic at 146.5[\( dB \)] rms SPL is clearly visible for richer operating conditions. Likewise, the same tendency is observable in the trade-off diagram \( NO_x \) versus SPL of MIC2 in Figure 6.21. The rise of the fuel concentration in the
regions of the injection ports causes a narrower stratification within the temperature zones, positively affecting flame stability characteristics but simultaneously forcing the potential of NO\textsubscript{x} formation by higher top level temperatures of the combustion process. However, NO\textsubscript{x} levels show a steeper slope for increasing rates of the secondary fuel injection for richer baseline operating conditions, whereby levels of the lean operating condition (\lambda\textsubscript{afr} = 2.4) are gently inclining. Similarly, the flame position \(PM_{\text{mean, normalized}}(1(0) = \text{at (far from) the dump plane of the EV burner) is attached closer to the burner exit with rising rates of the secondary fuel injection. The altered energy distribution of the pressure spectra as a consequence of the steady-state secondary fuel injection is illustrated in Figure 6.22 with an exemplary operating condition at \(T_{\text{mix}} = 700\,[K], \dot{m}_{\text{air}} = 36\,[g/s] \) and \(\lambda_{\text{afr}} = 2.0[\cdot].

The secondary gas flow
(steady) has already the ability without any controller to reduce the first longitudinal mode (−20[dB]) to an acceptable level by all means, if there would not be the severe drawback of the strong increased formation of $NO_x$ (+150%). Frequency bands higher than 1500[Hz] are less, or even not, affected by the steady gas flow through the two injection ports and thus not stated at this position.

### 6.4. Active instability control

The outcomes of the previous investigations with the actuator using the secondary injection system (in this case the two-hole configuration) set the main objectives and the deliverable of a control algorithm:

- Combustion instabilities are to be damped actively over a broad range of frequencies with a certain robustness of the feedback algorithm.
- New instabilities, as a result of the forcing of the reaction zone with the control algorithm, must be prevented.
- Emission levels, mainly nitrogen oxide ($NO_x$), should be negatively affected by the controller as less as possible.

However, the control algorithm implemented on the test rig remains restricted to simple phase-shift controllers and further advanced control designs went beyond the scope of this work. Control algorithms, such as robust $LQG/LTR$ and $H_{\infty}$-controller, require a model based design. Within the work of A. Niederberger et al. ([68],[69]), a physical model is developed and various controllers are derived and implemented in the same test rig using the identical fuel actuator with the two port injection. Furthermore,

**Figure 6.22.** Pressure spectra for different stationary gas flow rates through the secondary fuel injection system (two-hole configuration) with no forcing frequency and no amplitude for a fixed operating condition $T_{mix} = 700[K]$, $m_{air} = 36[g/s]$ and $\lambda_{air} = 2.0$. 
Figure 6.23. Pressure spectra (SPL of MIC2) for different operating conditions stabilized with a phase-shift controller using the secondary fuel injection system as an actuator.

Top: $\dot{m}_{\text{air}} = 30\, \text{g/s}$, $T_{\text{mix}} = 600\, \text{[K]}$ and $\lambda_{\text{air}} = 1.8$.

Middle: $\dot{m}_{\text{air}} = 36\, \text{g/s}$, $T_{\text{mix}} = 700\, \text{[K]}$ and $\lambda_{\text{air}} = 2.0$.

Bottom: $\dot{m}_{\text{air}} = 40\, \text{g/s}$, $T_{\text{mix}} = 700\, \text{[K]}$ and $\lambda_{\text{air}} = 2.2$. 

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his work compares the performances of the different control algorithm (including a controller using a genetic algorithm) and its impact on the acoustic behaviour of the combustor. From this, it can be stated that smarter controllers influence flame stabilization in a more favourable way as the results presented here which are obtained with a phase-shift controller and therefore fulfil the above listed conditions better and better (e.g. see the controller-oriented work of Niederberger [68]).

6.4.1. Performance of the phase-shift controller

The phase-shift controller (closed-loop control system) was implemented in a state-of-the-art real time system (dSPACE) according to Niederberger [68]. The pressure signal in the downstream section of the combustor (MIC1) was used as a feedback signal and filtered before sending the signal to the phase-shift controller. The following parameters of the phase-shift controller were individually adjusted for every investigated operating condition:

The command signal for the GMM valve was multiplied with a controller gain $G_{controller}$ to adjust the gas flow modulation and delayed by a time lag element $\Delta \tau_{controller}$ to adapt the correct phasing of the valve commanded secondary gas flow rate in order to suppress heat release oscillations in the combustion chamber. In addition, a controller offset $O_{controller}$ was added to the signal to set the steady-state gas flow rate through the GMM valve and the secondary injection ports whereby $O_{controller}$ influences the available control authority of the GMM valve. For all conditions, the main gas flow rate was reduced by the gas flow fed through the secondary injection system to guarantee a constant global air/fuel equivalence ratio $\lambda_{afr}$.

Pressure spectra of the microphone signal measured close at the dump plane (MIC2) are shown in Figure 6.23 for the baseline case (no secondary flow), for a condition with a controller offset $O_{controller}$ (steady-state secondary gas flow) and for a controlled operating condition using the phase-shift controller with the identical controller offset $O_{controller}$ (modulated secondary gas flow). For instable flames (see Figure 6.23, spectra at the top) with large pressure oscillations (flame type $\Delta$, $\Delta'$), the phase-shift controller is able to reduce the first longitudinal mode up to $-45[dB]$ and even the noise floor is shifted to lower levels ($\approx -10[dB]$), which results in an amazing reduction of the rms sound pressure level from $166[dB]$ down to $147[dB]$!

A drawback in terms of frequency content of the signal (beside these outstanding results obtained with this control scheme), is the generation of a new peak in the spectra on the lower frequency range ($<45[Hz]$) due to a slight untimely and/or unfitting secondary injection. However for less unstable $(\Delta a)$, $(\Delta r)$ operating conditions, the performance of the phase-shift controller is significantly reduced and completely disappeared for very lean operating conditions (flame types $(\Delta)$ and $(\Delta r)$).
6.4.2. Controller effects on the reaction zone

To investigate the impact of the controlled secondary injection flow and the resulting stabilized reaction zone, time-averaged intensity images of the OH chemiluminescence were acquired for several operating conditions. Due to the fact that the implementation of the phase-shift controller results in such a stabilized and silent flame, a characteristic dominant frequency resulting from pressure waves or fluctuations of the flame position (photomultiplier) is absent in the combustion chamber and therefore, no phase-averaged imaging technique is possible for the instability controlled flame (missing triggering reference).

Intensity images of the untriggered OH chemiluminescence are shown for two different operating conditions (Figure 6.24 and Figure 6.25) in each case comparing the

![Time-averaged OH chemiluminescence intensity images for the baseline](image)

Figure 6.24. Time-averaged OH chemiluminescence intensity images for the baseline $\dot{m}_{inj} = 30\text{g/s}$, $T_{inj} = 600\text{K}$ and $\kappa_{sd} = 1.8$ (left) and the stabilized flame with a phase-shift controller (right). For the top row, the intensity of images is normalized within a single image, whereas for the row at the bottom, images are normalized with each other (baseline and controlled image).
baseline condition (no secondary injection) with the phase-shift controlled flame. The unstable flame, corresponding to the slow transition of flame type (iii) was operated at $T_{\text{mix}} = 600[\text{K}]$, $\dot{m}_{\text{air}} = 30[\text{g/s}]$, $\lambda_{\text{air}} = 1.8[-]$ and phase-shift controlled with a secondary mass flow rate of $\dot{m}_{\text{stab}} / \dot{m}_{\text{gas,tot}} = 11[\%]$. The upper row of Figure 6.24 shows images, where the intensity is normalized within each individual image, then again at the lower row, images were normalized with each other to compare the differences in the intensity value.

In the case where only untriggered images were taking into account, unstable flames produce smereed intensity images by their movement within a period of instability (see image on the lower left of Figure 6.24). In contrast, controller stabilized flames posses smaller fluctuations of the flame position and therefore exhibit a more compact reaction zone than unstable flames. Although the flame structure of the baseline (flame type (ii)) pertains stronger to the flame type (iii) the controlled cases than to all other investigated flame types, a correlation to this flame type with an existing “natural-appeared” flame structure remains difficult. On no baseline condition, the tendency to build a “heart-shaped” reaction zone was observed, but in return, the central feature of compacter flame shapes along with an increased flame temperature is identical. However, the influence of the controller on the flame shape where the baseline is already stable, is marginal (see Figure 6.25) and the flame has only a somewhat consolidated reaction zone (more heart-shaped).

Likewise, when it was attempted to stabilize reaction zones of flame type (iii) by the phase-shift controller, the position of the flame front remains almost unaltered (Figure 6.26, right). Conversely, the responsiveness of leaner flame structures (flame type (i) and (ii)) is somehow different compared to flame type (iii) (and (iv)). Although
the influence of the phase-shift controller is evanescent in terms of a potential to reduce pressure oscillations in the combustor, the position of the flame front is strongly affected and the whole reaction zone is stabilized closer at the exit of the burner (Figure 6.26, left). Since for this cases (steady-state gas flow (O) and actively controlled gas flow (C)) the mean gas flow through the secondary fuel injection system is held constant as well as all other global parameters to operate the test rig, the augmented influence to pull the reaction zone near the dump plane is an exclusively nature of the controller.

Totally reversed is the characteristics of the flame front for controller-stabilized flames exhibiting a flame structure within the slow transition of flame type \( \text{(ib)} \) (see Figure 6.27, left). In contrast to previous observations for other flame types, the flame front of a controller-stabilized flame is located farther downstream as its position of the baseline case (flame type \( \text{(ib)} \)).

Moreover, if the positions of the flame front obtained at different phase angles during one cycle of the instability were taken into account, the flame front of the controlled flame is stabilized at the same position at its farthest of the unstable flame (Figure 6.27). The shape of the flame front of the controller-stabilized flame looks similar to the shape obtained at a phase angle of 180° (No. 5) and is even more axis-symmetric regarding to the \( y \) axis of the combustor as the shape of flame fronts of unstable baselines. This is an astonishing fact, especially since the tendency to cause asymmetric flame shapes due to the secondary injection was found for forced reaction zones and pointed out in section 6.3 (see Figure 6.16 on page 139).
Figure 6.27. Position of the flame front for a constant operating condition of $u_{air} = 30$ [g/s], $T_{mix} = 600$ [K] and $\lambda_{air} = 1.8$. Left: Position of the flame front derived from the untriggered OH chemiluminescence intensity images showing the baseline (B), the influence of the stationary secondary gas flow (O) using the same amount as the controller requires and the phase-shift controlled flame (C). Right: Position of the flame front obtained for the investigated phase angles of the unstable flame (numbers 1 to 8) and the stabilized front of the controlled flame (C).

6.4.3. Influence of the operating condition on the controller performance

For an appraisement of the functional efficiency of the secondary injection strategy (including the phase-shift controller) several tests were performed covering all relevant flame structures at different operating conditions. Parameters of the controller were individually tuned to achieve the best compromise between an overall reduction of the sound pressure level in the combustion chamber in conjunction with maintaining the formation of nitrogen oxide ($NO_x$) at a low level.

In doing so, the gas mass flow rate used by the controller and the rate which responds only to the mean command offset of the GMM valve are not equal any more, unlike in the foregoing results. This is traced back to the fact that the valve is not operated in the linear regime of the characteristic curve of the mass flow rate of the GMM valve (see Figure 6.5). In the lower non-linear region, for instance, a sine wave oscillation around a mean command voltage results in an increased averaged mass flow rate distinct from the mass flow rate originating only from the similar mean command voltage without any superimposed oscillation.

However, operating the valve in this for the actuator suboptimal regime outyields lower $NO_x$-levels by giving up parts of the control authority. The differences of the gas mass flow rates conducted through the secondary injection as a cause of the phase-shift controller and of the command offset are depicted in Figure 6.28 for test points, to check the controller performance.
Active instability control

Figure 6.28. Secondary fuel injection rates $m_{valve}$ shown as a fraction of the total gas flow rate $m_{gas, tot}$ for several operating conditions (including different flame types).

A summary is presented in Figure 6.29, showing the sound pressure level SPL of MIC2 and the $NO_x$-levels for the baseline case as well as for the steady-state flow through the valve at the corresponding command offset used by the controller and for the case where a phase-shift controller is turned on.

The controller is able to reduce sound pressure level to a greater or lesser extent for all operating conditions including stable and unstable flames. In particular for very noisy operating conditions already at the baseline, the controller exhibits an excellent poten-
tial to reduce pressure oscillations (A maximum reduction of rms SPL of $-18\,[dB]$) by mainly affecting fundamental oscillations in the combustion system (A maximum peak to peak reduction of $-45\,[dB]$ SPL). The minimal achieved sound level in the combustor obtained with the phase-shift controller lies close to the asymptotic value of 146.5\,[dB] rms SPL derived in chapter 6.3.3.

Any implementation of a controller with this secondary fuel injection causes a slight increase in the $NO_x$-levels (Figure 6.30). For the reaction zone of the flame type IIIa, the rise of the $NO_x$-levels is unjustifiable considering the small reduction of the sound pressure level due to the phase-shift controller. For the weak flame type of IIb), the controller has no explicit influence and produces no additional $NO_x$ emissions. Both flame types, IIb) and IIIa), take no advantage of the secondary fuel control and it can therefore just be omitted as well.

For the flame type of IIb) and IIIb), sound pressure levels of the controlled flame are well below the baseline and the drawback of risen $NO_x$-levels is marginal. In this domain, the secondary fuel injection and the phase-shift controller are advisable and are able to stabilize thermoacoustic instabilities in extenso.
Active instability control
7 Conclusions and Outlook

7.1. Flame properties

The flame structure of an atmospheric, swirl stabilized burner fired with natural gas has been investigated as a function of total mass flow rate, preheat temperature, air/fuel equivalence ratio and combustion chamber length. This investigation revealed the existence of several flame states, ranging from “weakly burning” flames, where combustion processes only took place further downstream in the central recirculation zone, to “strongly burning” flames where the flame extended into the outer recirculation zone, the borders of which were linked to computed adiabatic flame temperatures.

Two distinct transitions playing a determining role for the acoustic properties of the flame have been identified: a fast transition, by which the flame position and shape abruptly changed, following a very small variation in the air/fuel equivalence ratio; and a slow transition where the flame properties varied smoothly and continuously over a narrow band of air/fuel equivalence ratios. The slow transition coincided with the maximum pressure drop across the flame and with the largest pressure oscillations within the combustion chamber. OH-chemiluminescence measurements one diameter downstream of the burner, on the axis of symmetry, revealed a two-step transition from weakly to strongly burning flames.

Acoustic measurements identified several unstable thermoacoustic modes with frequencies ranging from 200 [Hz] for the dominant mode up to several kHz for high frequency modes. It was established that two modes at Strouhal numbers of $St = 0.3$ and $St = 1.5$ were independent of both the temperature and the length of the combustion chamber, but scaled with the mean velocity at the burner outlet. These two modes were present for all investigated parameters, including non-reacting flows, and had a constant normalized frequency (Strouhal number). Other excited frequency bands have been identified as acoustic eigenmodes of the combustion chamber and were found to scale with the speed of sound.
Control characteristics

The normalized contribution of main unstable modes was calculated as a function of air/fuel equivalence ratios. This confirmed that the maximum acoustic oscillations were reached for axially stretched flames (slow transition) and additionally showed that the mode associated with the 1st longitudinal acoustic mode was consuming most of the energy (long combustion chamber). On the contrary, two modes existed simultaneously and consumed most of the energy during operation with the short combustion chamber. In addition, the normalized contribution of high frequency modes increased globally for very rich and very lean conditions.

In order to visualize the characteristics of the flame along with thermoacoustic instabilities, phase-averaged OH chemiluminescence pictures were systematically acquired to investigate the influence of the air/fuel equivalence ratio and the mixture temperature, as well as the air mass flow rate on the investigated flame shapes. In particular, the different intensity levels of the reaction zone, the flame front, the position of the flame front, and its movement have been analysed.

The role of air/fuel equivalence ratio fluctuations, and possible variations of the power density of the flow rate, as driving mechanisms of thermoacoustic instabilities for the investigated gas turbine combustor, may be disregarded on the basis of the results from the methane absorption and acetone-PLIF measurements in the upstream and downstream sections of this combustor.

7.2. Control characteristics

The secondary fuel injection and phase-shift controller are a potential system to stabilize thermoacoustic instabilities in gas turbine combustors. The uncoupling of heat release rate oscillations, velocity/pressure fluctuations, burnt/unburnt reactant gas mixing rates are directly achieved by introducing controlled air/fuel equivalence ratio fluctuations, through which the instability is significantly reduced or even completely stabilized.

The control authority obtained by a secondary fuel injection is quite adequate to control combustion instabilities, dramatically reducing sound pressure levels within the combustor. The maximum reduction of the average (rms) sound pressure level by 18 dB was achieved mainly by affecting fundamental oscillations in the combustion system, such that a maximum reduction of the peak sound pressure level of 45 dB resulted.

It has been shown that any intervention by the controller with the secondary fuel injection causes a slight increase (maximum of $2[ppmv]_{dry}$) in the $NO_x$-levels. In either case, a stratification of the temperature zone with a secondary injection technique always
tends to result in more stable operating conditions and results in an increased nitrogen oxide formation potential. The extent of an increased formation of \( NO_x \), however, is marginal compared to the possible reduction of sound pressure levels attainable by optimising the injection parameters used for the stabilization of the lab-scale combustor. When applying the proposed secondary injection system to the real gas turbine combustor (EV17, EV19 and AEV19), the drawback of higher \( NO_x \)-formation rates can be excluded, as the main gas feed line of the real gas turbine combustor injects at the same position in the flow field as on the EV combustor.

In order to go deeply into the question, “How to improve the secondary fuel injection strategy?”, it seems promising to use liquid fuels for the secondary fuel flow. The actuator operated with liquids exhibits a favourable transfer function with a flow resonant frequency close to the mechanical response of the rod, which improves the ability to control modes at even higher frequencies. However, additional mixing and evaporation times, possibly affecting the band of controllable frequencies negatively, must be taken into consideration to prevent increased emissions (e.g. \( NO_x \) and soot) when using such a liquid fuel injection strategy.

Besides basic tests of the fuel flexibility of the control system, a further investigation should cover the application of hydrogen enriched gas as a fuel used by the secondary injection system. Small amounts of hydrogen (or hydrogen rich gas) introduced by the secondary injection system flow may alter the flame structure, as well as the time scales of the combustion process. Provided that both effects are accurately applied, this policy will result in a damped feedback process.
A Appendix

A.1. Measuring instruments

A.1.1. Non-contact displacement measuring system (gap sensor)

<table>
<thead>
<tr>
<th>Specifications (Model: AEC-5503A / PU-03A)</th>
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<tr>
<td>Output voltage</td>
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<td>Measuring range</td>
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<td>Frequency characteristics</td>
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<td>Resolution</td>
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<td>Temperature range</td>
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<td>Thermal characteristics</td>
</tr>
<tr>
<td>Power supply</td>
</tr>
</tbody>
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Table A.1. Gap sensor of the GMA prototype valve

A.1.2. High precision power amplifier (NF 4505)

<table>
<thead>
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<th>Specifications (Model: NF Corporation; NF4505)</th>
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<tbody>
<tr>
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<tr>
<td>Rated output voltage</td>
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<td>Maximum output voltage</td>
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<tr>
<td>Gain in constant voltage mode (CV)</td>
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<tr>
<td>Rated output power</td>
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<tr>
<td>Peak current</td>
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<tr>
<td>Rated output voltage</td>
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<td>Frequency response</td>
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<td>Harmonic distortion</td>
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Table A.2. High precision power amplifier for the GMA prototype valve
A.1.3. Dynamic pressure measuring devices

A.1.3.1. Free-field ¼-inch Microphone

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<th>Specifications (Model: Brüel &amp; Kjær, Type 4939)</th>
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<td>Polarization Voltage</td>
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<td>Frequency Response</td>
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<td>Free-field Response 4 Hz to 100 kHz</td>
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<td>Lower Limiting Frequency (–3 dB)</td>
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<tr>
<td>Diaphragm Resonance Frequency</td>
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<tr>
<td>Capacitance (Polarized 250 Hz)</td>
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<td>Upper Limit of Dynamic Range (3% Distortion)</td>
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<tr>
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<td>Operating Humidity Range</td>
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<tr>
<td>Influence of Humidity</td>
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<tr>
<td>Vibration Sensitivity (&lt;1000 Hz)</td>
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Table A.3. Specifications of the microphone head

Figure A.1. Typical frequency response of a microphone used for the current investigation. Every microphone were individually calibrated and has an typical response characteristic.
### A.1.3.2. Falcon Range ¼-inch Microphone Preamplifier

<table>
<thead>
<tr>
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<td><strong>Input impedance</strong></td>
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<td><strong>Output impedance</strong></td>
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<tr>
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Table A.4. Specifications of the microphone preamplifiers

### A.1.3.3. 4-channel Microphone Power Supply

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</tr>
<tr>
<td></td>
</tr>
<tr>
<td><strong>Amplification</strong></td>
</tr>
<tr>
<td><strong>Noise level</strong></td>
</tr>
<tr>
<td><strong>Power:</strong></td>
</tr>
</tbody>
</table>

Table A.5. Power supply used in conjunction with the microphone preamplifiers

### A.1.3.4. Sound level calibrator

<table>
<thead>
<tr>
<th>Specifications (Model: Brüel&amp;Kjær, Type 4231)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Nominal sound pressure level</strong></td>
</tr>
<tr>
<td><strong>Frequency</strong></td>
</tr>
<tr>
<td><strong>Equivalent free-field level</strong></td>
</tr>
<tr>
<td><strong>Equivalent random incident level</strong></td>
</tr>
<tr>
<td><strong>Total harmonic distortion (THD)</strong></td>
</tr>
<tr>
<td><strong>Level stability</strong></td>
</tr>
</tbody>
</table>

Table A.6. Specifications of the battery operated sound level calibrator
Measuring instruments

A.1.4. Photosensor module

<table>
<thead>
<tr>
<th>Specifications (Model: Hamamatsu, Type H5783-03)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spectral response</td>
</tr>
<tr>
<td>Effective area</td>
</tr>
<tr>
<td>Peak sensitivity wavelength</td>
</tr>
<tr>
<td>Luminous sensitivity at the cathode (min: 40 μA/lm; typ: 70 μA/lm)</td>
</tr>
<tr>
<td>Luminous sensitivity at the anode (min: 10 μA/lm; typ: 50 μA/lm)</td>
</tr>
<tr>
<td>Rise time</td>
</tr>
<tr>
<td>Control voltage adjustment range</td>
</tr>
<tr>
<td>Operating temperature</td>
</tr>
</tbody>
</table>

Table A.7. Specifications of the photosensor module housing a metal package PMT and a high voltage power supply circuit

![Cathode radiant sensitivity of the photosensor module](image)

Figure A.2. Cathode radiant sensitivity of the photosensor module

A.1.5. Infrared photovoltaic detector

<table>
<thead>
<tr>
<th>Specifications (Model: Vigo InAs, Type PVI-2TE-InAs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>λ_{peak} : λ_{cutoff}</td>
</tr>
<tr>
<td>Detectivity at λ_{peak}</td>
</tr>
<tr>
<td>Responsivity</td>
</tr>
<tr>
<td>Resistance</td>
</tr>
<tr>
<td>Response time</td>
</tr>
<tr>
<td>Field of view</td>
</tr>
<tr>
<td>Operating temperature</td>
</tr>
</tbody>
</table>

Table A.8. Specifications of the infrared photovoltaic InAs detector
A.2. Additional theory of acoustics

A.2.1. The sinusoidal sound wave

A sinusoidal function $\tilde{p}(t)$ is defined as:

$$\tilde{p}(t) = p_0 \cdot \sin(\omega t + \phi)$$

where $p_0$ stands for the pressure amplitude expressed in Pascal ([$Pa$]), $\omega$ for the angular frequency in [rad/s] and $\phi$ for the phase shift angle in [rad]. The wavelength of the sine [m] is given with

$$\lambda = \frac{c}{f}$$

The speed of sound $c$ in units of [m/s] can be written as $c = \sqrt{\gamma R_T}$ (in [m/s]), $R_T$ in [Pa m^3/kg], $\gamma$ = c_p/c_v. Hertz is the unit of the frequency $f$ of the wave and is equal to $\frac{1}{[s]}$.

A.2.2. The Root Mean Square (RMS) of the pressure signal

To obtain the amplitude of a general input signal $\tilde{p}(t)$, the use of the root mean square (RMS) pressure $p_{rms}$ come into considerations.

$$\tilde{p}_{rms} = \sqrt{\frac{1}{T} \int_{(t-T)}^{t} \tilde{p}(t)^2 \, dt}$$

In terms of the period $T$, the corresponding frequency is given by $f = \frac{1}{T} = \frac{\omega}{2\pi}$.

For discrete (sampled) signals, the root mean squared signal is represented by:

Figure A.3. Spectral response of the InAs detector
Assuming the signal is composed of pure sinusoids, the following relationship between the amplitude $p_0$ and the RMS value is valid:

$$p_{rms} = \frac{p_0}{\sqrt{2}}$$  \hspace{1cm} (A.5)

### A.2.3. The decibel scale (dB)

The threshold of human hearing is at $p_{th,\text{rms}} = 20 \mu Pa$ and at a level of seven orders of magnitude higher the onset of hearing damage is at about $p_{tp,\text{rms}} = 20 mPa$. Hence, to represent the wide hearing range and to make further calculations simpler, the decibels, abbr. dB (-) as a logarithmic scale is used.

Generally, decibels define a ratio between two levels:

$$\Delta x = 20 \log_{10} \left( \frac{x_1}{x_2} \right) \text{ dB for linear values } x_1, x_2$$ \hspace{1cm} (A.6)

$$\Delta x = 10 \log_{10} \left( \frac{x_1}{x_2} \right) \text{ dB for quadratic values } x_1, x_2$$ \hspace{1cm} (A.7)

+ 20dB represents a multiplication factor of 10, + 6 dB represents a multiplication factor of 2 and + 3 dB correspond to a noticeable difference in hearing.

### A.2.4. Sound Pressure Level (SPL)

The definition of the sound pressure level, SPL is

$$L_p = 20 \log_{10} \left( \frac{p_{rms}}{p_{ref}} \right) \text{ dB}$$  \hspace{1cm} (A.8)

$L_p$ or $SPL$ as an abbreviation of sound pressure level are similar used! The reference pressure $p_{ref}$ used as basis for the decibel scale is defined by the pressure waves for the threshold of hearing $p_{ref} = 20 \mu Pa$. 

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A.3. Additional results

A.3.1. Analytical eigenmode analysis (Bessel function)

The Helmholtz's equation, as a specific form of the homogeneous wave equation is usually defined as (where $k$ is the wave number):

$$\nabla^2 \phi + k^2 \phi = 0$$  \hspace{.5cm} (A.9)

The Helmholtz's equation can be written explicitly for cylindrical coordinates ($r, \theta, z$) in the form of (evaluation of $\nabla^2$ in cylindrical coordinates)

$$\nabla^2 \phi = \frac{\delta^2 \phi}{\delta r^2} + \frac{1}{r} \frac{\delta \phi}{\delta r} + \frac{1}{r^2} \frac{\delta^2 \phi}{\delta \theta^2} + \frac{\delta^2 \phi}{\delta z^2} = -k^2 \phi$$  \hspace{.5cm} (A.10)

With a substitution of the form $\phi = Z(z)R(r)\Theta(\theta)$, it is possible to successfully separate the variables of equation A.10.

$$ZR'' \Theta + \frac{1}{r} ZR' \Theta + \frac{1}{r^2} ZR \Theta'' + Z'' R \Theta = k^2 Z R \Theta$$ \hspace{.5cm} (A.11)

$Z$ is firstly separated by putting all the $z$-dependence in one term, so that $Z''/Z$ can only be a constant, taken as $-s^2$.

$$\frac{R''}{R} + \frac{1}{r} \frac{R'}{R} + \frac{1}{r^2} \frac{\Theta''}{\Theta} + \frac{Z''}{Z} = -k^2 \quad \rightarrow \quad \frac{Z''}{Z} = -s^2$$  \hspace{.5cm} (A.12)

In a similar way, the separation of $\frac{\Theta''}{\Theta}$, which has all the $\theta$ dependence, is another constant $-n^2$, again for convenience.

$$r^2 \left[ \frac{R''}{R} + \frac{1}{r} \frac{R'}{R} + \left( k^2 - s^2 \right) \right] + \frac{\Theta''}{\Theta} = 0 \quad \rightarrow \quad \frac{\Theta''}{\Theta} = -n^2$$  \hspace{.5cm} (A.13)

Combining equation A.12 and A.13 gives

$$R'' + \frac{1}{r} R' + \left[ k^2 - s^2 - \frac{n^2}{r^2} \right] R = 0$$  \hspace{.5cm} (A.14)

Finally, the equation A.14 for $R(r)$ and $\Theta(\theta)$ remains (Bessel's equation), so that $R$ is a linear combination of $J_n$ and $Y_n$ of the well-know solution

$$\Theta(\theta) = A \cos(n\theta) + B \sin(n\theta)$$  \hspace{.5cm} (A.15)

$$R(r) = A \cdot J_n(kr) + B \cdot Y_n(kr)$$  \hspace{.5cm} (A.16)

If the shape is specified as a cylinder with radius $r_0$, the appropriate solutions are of the form
\[ \phi = J_n(kr) \cos(n\theta) \text{ or } \phi = J_n(kr) \sin(n\theta) \]  
(A.17)

The boundary condition at the rigid surface of the cylinder is that the normal velocity must vanish, which gives an additional formulation of equation A.17.

\[ \left[ \frac{dJ_n(kr)}{dr} \right]_{r=r_0} = 0 \]  
(A.18)

Finally, the several roots \( \beta_{mn} \) of the Bessel function \( J_n(kr) \) are the solution of the eigenvalue problem stated in equation A.9.

Distinct values of the roots correspond to different eigenvalues \( \lambda_{mn} \), or with the help of equation A.19, can be transformed to the characteristic azimuthal and radial modes \( f_{nm} \) of the acoustic, cylindrical problem.

\[ \lambda_{mn} = \left( \frac{\beta_{mn}}{r_0} \right)^2 \rightarrow f_{mn} = \frac{c}{2\pi r_0} \beta_{mn} \]  
(A.19)

<table>
<thead>
<tr>
<th>( \beta_{mn} )</th>
<th>Radial Modes (n)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Azimuthal Modes (m)</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>1.841</td>
</tr>
<tr>
<td>3</td>
<td>4.201</td>
</tr>
<tr>
<td>4</td>
<td>5.317</td>
</tr>
<tr>
<td>5</td>
<td>6.415</td>
</tr>
</tbody>
</table>

Table A.9. Roots of the Bessel function \( J_n(kr) \)
A.3.2. Eigenmode analysis with FEMLAB

![Eigenmode analysis results](image)

*Figure A.4. Some results of the FEMLAB eigenmode analysis*

A.3.3. Emissions

![Emissions results](image)

*Figure A.5. Upper figure: Flame structure for the long combustion chamber (LCC) for a constant mass flow rate $\dot{m}_{\text{air}} = 36 \text{[kg/s]}$ (left) and $O_2$ values in the $(\lambda_{\text{air}}, T_{\text{mix}})$ plane (right). Lower figure: The conditions are the same as above and the $CO_2$ emissions are shown at a dry basis (left) and at a reference of $15\% - O_2$ (right).*
Figure A.6. Left figures: CO, UHC and NO\textsubscript{\textsc{s}} emissions shown at a dry basis in the ($\lambda_{\text{eff}}, T_{\text{mix}}$) plane for a constant mass flow rate $\dot{m}_{\text{in}} = 36$[kg/s]. Right figures: Emissions are shown at a reference of $15\%\text{-}O_2$ (same conditions as left).
Figure A.7. Iso-contours of the pressure amplitude at \( \dot{m}_{air} = 361 \text{ g/s} \) (LCC, MIC1, variation of the mixture temperature)
Figure A.8. Iso-contours of the pressure amplitude at $\dot{m}_{\text{in}} = 35\text{[g/s]}$ (LCC, MIC2, variation of the mixture temperature)
Figure A.9. Iso-contours of the pressure amplitude at $u_{in} = 36[g/s]$ (LCC, MIC3, variation of the mixture temperature)
Figure A.10. Iso-contours of the pressure amplitude at $T_{\text{mix}} = 700[K]$ (LCC, MIC2, variation of the air mass flow rates)
A.3.5. OH chemiluminescence images of the SCC

Figure A.11. Time-averaged images of the OH chemiluminescence intensity investigated in the short combustion chamber (SCC).
Top: Flame type (a) at \( \lambda_{\text{eff}} = 2.8 \), \( T_{\text{init}} = 700 [K] \), \( \dot{m}_{\text{air}} = 40 [g/s] \).
Middle: Flame type (b) at \( \lambda_{\text{eff}} = 2.6 \), \( T_{\text{init}} = 700 [K] \), \( \dot{m}_{\text{air}} = 40 [g/s] \).
Bottom: Flame type (c) at \( \lambda_{\text{eff}} = 2.4 \), \( T_{\text{init}} = 700 [K] \), \( \dot{m}_{\text{air}} = 40 [g/s] \).
Figure A.12. Time-averaged images of the OH chemiluminescence intensity investigated in the short combustion chamber (SCC).

Top: Flame type ( tipo ) at $\lambda_{\text{eff}} = 2.2$, $T_{\text{max}} = 700\,^\circ\text{C}$, $m_{\text{fuel}} = 40\,\text{g/s}$.

Middle: Flame type ( tipo ) at $\lambda_{\text{eff}} = 2.0$, $T_{\text{max}} = 700\,^\circ\text{C}$, $m_{\text{fuel}} = 40\,\text{g/s}$.

Bottom: Flame type ( tipo ) at $\lambda_{\text{eff}} = 1.8$, $T_{\text{max}} = 700\,^\circ\text{C}$, $m_{\text{fuel}} = 40\,\text{g/s}$.
Figure A.13. Time-averaged images of the OH chemiluminescence intensity for the short combustion chamber (SCC) according to Figure 5.1 ($m_{\text{fuel}} = 40\text{g/s}$).
Figure A.14. Time-averaged images of the OH chemiluminescence intensity (normalized over all images) for the SCC according to Figure 5.1 ($m_{\text{air}} = 401g/s$).
Figure A.15. Time-averaged images of the OH chemiluminescence and the OH-PLIF intensity whereby in the second and the fourth column the images listed are normalized within all images) for the LCC. Operating conditions: $\lambda_{eff} = 1.8$, $\tau_{mix} = 600[\kappa]$ and $m_{fuel} = [30, 36, 42, 48][g/l]$ from the bottom up.
**Figure A.16.** Probability of the flame front location derived from phase-averaged OH chemiluminescence and OH-PLIF intensity images for an operating condition of $\lambda_{air} = 1.8$, $T_{air} = 600[K]$ and $\dot{m}_{air} = 30[g/s]$ (LCC).
A.3.6. Acetone LIF applied in a free jet modulated with a the GMM valve

Figure A.17. Phase-averaged images of the acetone LIF intensity of the free jet. Operating condition of GMM valve: $f_{\text{forcing}} = 100 \, \text{Hz}$, $A_{\text{forcing}} = 0.3V$ and $O_{\text{forcing}} = 0.3V$.

Figure A.18. (Part I) Phase-averaged images of the acetone LIF intensity of the free jet. Operating condition of GMM valve: $f_{\text{forcing}} = 200 \, \text{Hz}$, $A_{\text{forcing}} = 0.3V$ and $O_{\text{forcing}} = 0.3V$. 
Figure A.19. (Part II) Phase-averaged images of the acetone LIF intensity of the free jet. Operating condition of GMM valve: \( f_{\text{forcing}} = 200 [Hz], \ A_{\text{forcing}} = 0.3 |V| \) and \( \phi_{\text{forcing}} = 0.3 |V| \).

Figure A.20. Phase-averaged images of the acetone LIF intensity of the free jet. Operating condition of GMM valve: \( f_{\text{forcing}} = 500 [Hz], \ A_{\text{forcing}} = 0.3 |V| \) and \( \phi_{\text{forcing}} = 0.3 |V| \).
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References


References


References


and


Curriculum Vitae

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December 5, 2005, examination for a doctorate