

Effect of forced equivalence ratio fluctuations combined with hydrogen enrichment in a lean technically pre-mixed (swirl-stabilised) methane combustor

Master Thesis

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EFFECT OF FORCED EQUIVALENCE RATIO FLUCTUATIONS COMBINED WITH HYDROGEN ENRICHMENT IN A LEAN TECHNICALLY PRE-MIXED (SWIRL-STABILISED) METHANE COMBUSTOR

MASTER THESIS

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PREFACE

This report presents the work for obtaining my master in aerospace engineering and marks the culmination of my 7 years in Delft. The topic of my thesis, forced fluctuations on hydrogen-enriched methane flames, saw me explore a wide range of topics, requiring me to rely on the knowledge and skills I obtained throughout these 7 years. Especially, the fact that it was an experimental thesis required me to delve in fields that I hadn't encountered since the beginning of my bachelor or that were completely new for me such as the Labview code for data acquisition. It was a very interesting thesis topic for myself, but challenging. I learnt a lot, also about myself, during the thesis and will take the lessons learnt with me. I believe that the finalisation of this thesis and my masters marks a nice end point for my years in Delft and finalisation the beginning of a new chapter in my life. A chapter where I will continue to pursue growth and learning.

The report is quite extensive, therefore I suggest the reader to only focus on the chapters and sections that interest them. The literature study is very broad, thorough and even delves in some fundamental aspects as I wanted to lay a good foundation at the beginning of my research. The outline gives insight into the research question and objective. The set-up section is also quite detailed as I needed to create a good picture for myself in order to identify possible disturbances and influences. The methodology section sheds light on the test campaigns and test matrix. It also provides insights on the details on the used measurement techniques. The experimental results chapter presents the experimental analysis and is followed by the numerical exploration of the topic. Conclusions are presented in the conclusion together with the recommendations.

I would also like to thank several that have been important during the writing of this thesis. First and foremost my supervisor Francesca for her help and availability. Secondly, I would to thank Sarah for her help during her entire thesis, especially during the experimental campaigns which wouldn't have been possible without her. I would also like the lab technicians, in particular Gert-jan, for helping me develop the bypass line with valve and in the case of Gert-jan its actuation. There is also Johannes who helped setting-up the system and make the measurements in the first two campaigns. Of course, I would also like to thank several people who were not directly involved with the thesis. Like my parents who have been very supportive and gave me the opportunity to even start in Delft. I would also like to credit the gang from the master room: Iacopo, Sara, Yoram and Guilia for their support. There is also Hortense, who was a great support during the final writing phase. Lastly I would like to thank my sister, grandparents, roommate Theo, Beatriz, Ian, Bart, Yannick, Victor, Noah and all my other friends and family for their support, motivation and helping me to take my mind off of things once in a while!

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SUMMARY

This master thesis explores the potential of hydrogen-enriched fuel blends in improving the performance and reducing emissions of lean premixed swirl-stabilized combustors in the aerospace sector. As industries face increased pressure to decarbonize amidst global energy crises and climate change, alternative fuels like hydrogen are being considered due to their lower carbon emissions compared to traditional hydrocarbon fuels.

By employing a high-speed solenoid flow valve to induce controlled fluctuations in the air-fuel mixture, the study investigates how these changes influence the overall behavior of the combustion process. The experimental setup for this thesis involved a swirl-stabilized multi-fuel combustor at the TU Delft Sustainable Propulsion Lab, capable of operating with methane and up to 100% hydrogen enrichment. The experimental analysis used a range of techniques, including Particle Image Velocimetry, SPOD and various acoustic and pressure measurements, to examine the impacts on the combustor's operation across different power settings and levels of hydrogen enrichment.

The findings reveal that adjusting the equivalence ratio through precise airflow modulation can significantly alter the unsteady combustion dynamics. Notably, this modulation enhances the stability of the flame by dampening the damaging thermoacoustic instabilities commonly associated with lean premixed combustors. Additionally, the research demonstrates that these adjustments can lead to changes in emission levels, notably in how nitrogen oxides and carbon dioxide are produced under varying conditions of hydrogen content and power settings.

A comprehensive numerical model was also developed to support and extend the experimental results, providing insights into the potential scalability and application of these findings to larger, real-world combustion systems. This model underscores the practical implications of the experimental findings, suggesting that such controlled fluctuations can be a viable method to optimize the design and operation of next-generation combustors.

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LIST OF SYMBOLS & ABBREVIATIONS

<i>Symbol</i>	<i>Meaning</i>	<i>Unit</i>	<i>Symbol</i>	<i>Meaning</i>	<i>Unit</i>
α	Thermal diffusivity	$\text{m}^2 \text{s}^{-1}$	Re	Reynolds number	–
β	Swirl decay rate	$\text{m}^2 \text{s}^{-1}$	Re_D	Diameter Reynolds number	–
δ	Velocity boundary layer	m	Re_t	Turbulent Reynolds number	–
δ^*	Velocity boundary layer displacement thickness	m	R_h	Inner radius swirl stream	m
δ_T	Thermal boundary layer thickness	m	R_n	Outer radius swirl stream	m
δ_f	Flame thickness	m	S_L	Laminar flame speed	cm s^{-1}
δ_q	Quenching distance	mm	S_T	Turbulent flame speed	cm s^{-1}
η	Charateristic length Kolmogrov scale	m	S_w	Swirl number	–
λ	Wavelength	$\text{m}^2 \text{s}^{-1}$	S_{w0}	Initial swirl number	–
ν	Momentum diffusivity	$\text{m}^2 \text{s}^{-1}$	St	Strouhal number	–
τ_0	Integral timescale	s	T_{ad}	Adiabatic flame temperature	K
τ_η	Kolmogrov timescale	s	u'_0	Turbulent fluctuations	m s^{-1}
τ_c	Chemical timescale	s	$\bar{v}$	Bulk velocity	m s^{-1}
ϕ_0	Swirler tip vane angle	°	v'	Turbulent fluctuations	m s^{-1}
ϕ	Equivalence Ratio	–	W	Channel width (diameter)	m
φ	Local vane angle	°	W_{eff}	Effective channel width	m
ψ	Blockage factor straight vanes	–	<i>Abbreviation</i>	<i>Meaning</i>	
χ	AAI percentage	–	AAI	Axial Air Injection	
ψ_{hel}	Blockage factor helical vanes	–	AIC	Active Instability Control	
A	Flame surface area	m^2	APPU	Advanced Propulsion & Power Unit	
A_T	Wrinkled flame surface area	m^2	APU	Auxiliary Power Unit	
A_{vane}	Projected vane area	m^2	BLI	Boundary Layer Ingestion	
D_{AB}	Mass diffusivity	$\text{m}^2 \text{s}^{-1}$	FB	Flash Back	
D_T	Turbulent diffusivity	$\text{m}^2 \text{s}^{-1}$	FFT	Fast Fourier Transform	
D	Molecular diffusivity	$\text{m}^2 \text{s}^{-1}$	FTF	Flame Transfer Function	
Da	Damköhler number	–	LES	Large-eddy simulation	
$f_{forcing}$	Forcing frequency	Hz	LHV	Lower Calorific Heating Value	
G_Θ	Axial flux of tangential momentum	kg m s^{-1}	HC	Hydrocarbons	
G_x	Axial flux of axial momentum	kg m s^{-1}	NOx	Nitrogen Oxides (NO and NO ₂)	
g_f	Critical velocity gradient	–	PLIF	Planar Laser Induced Fluoresence	
Ka	Karlovitz number	–	PIV	Particle Image Velocimetry	
Le	Lewis number	–	POD	Proper Orthogonal Decomposition	
L_e	Entrance length boundary layer growth	m	UV	Ultra Violet	
L_f	Characteristic length	m	(I)CCD	(Intensified) Charge-Coupled Device	
$\dot{m}_{AAI}$	AAI Swirler mass flow	kg/s			
$\dot{m}_{total}$	Total swirler mass flow	kg/s			
n_{vane}	Number of swirling vanes	–			
Pr	Prandtl number	–			
r_0	Outer radius	m			
$\bar{q}$	Mean heat release rate	W			
q'	Thermal fluctuations	W			

1

INTRODUCTION

Current global challenges, like climate change and the recent energy crises, put an immense pressure on the aerospace sector and hard-to-abate industries at large to accelerate their decarbonisation efforts and lower their emissions. [1, 2] In order to achieve this, these industries are looking to electrify their processes or to use more efficient combustion systems in combination with alternative low-carbon fuels. [3, 4] In recent years there has been a renewed interest in swirl-stabilised and lean premixed combustion systems given the improved efficiency and lower emissions associated with them. Before, major technical challenges regarding the increased susceptibility to develop combustion instabilities have held the development of lean premixed systems back. [5] However, the promise of more stringent regulations in the coming decades has renewed interest in lean combustion. [6, 7]

More recently, the research into lean premixed swirl stabilised systems has evolved to include alternative fuels, like hydrogen. Hydrogen is expected to play a crucial role in the energy transition as a versatile energy carrier and effective means of energy storage that complements renewable energy sources. [6] This role can be extended to combustion applications as its lack of carbon atoms mitigates the carbon emissions associated with traditional hydrocarbon fuels. That being said, there are still several technical challenges attributed to the higher burning velocity and adiabatic flame temperature that accompany hydrogen flames. These may cause the flame to flash back into the injector and increase NO_x emissions if not designed properly. Hydrogen's unique physical properties, like its high flame speed and wide flammability limits that result from its high diffusivity, make the hydrogen (enriched¹) flames behave fundamentally different from traditional, thoroughly researched hydrocarbon fuels. Beyond the challenges associated with its combustion, the world-wide supply chain of (clean) hydrogen still needs to develop and mature before hydrogen can be used as a stand alone fuel in any combustion application. Therefore, several research groups are focusing on the development of multi-fuel combustors that are able to switch from traditional hydrocarbon fuels, to hydrogen-enriched fuel blends, to pure hydrogen. The sustainable aircraft propulsion lab at the TU Delft is one of these research entities and intends to develop a lean premixed swirl stabilised hydrogen enriched methane combustor as a first step towards multi-fuel combustion under its APPU project [8].

However, this multi-fuel usage brings forth an additional technical challenge concerning the mitigation of combustion instabilities, such as thermoacoustic instabilities and flashback, that typically plague lean premixed combustors. [5, 9] Specifically, the various fuel blend ratios change the dynamics and structure of the flame significantly, which strongly influences the nature and behaviour of the combustion instabilities. This poses a problem as the frequencies and occurrence of the thermoacoustic instabilities change for the different fuel blends. Traditionally, these frequencies are dampened and suppressed by passive systems that only have a limited effective frequency range meaning that to accommodate this broader range the engine weight will have to increase. Similarly, the broader range of flame speeds over the different fuel blends makes it harder to design against flashback. To add to the difficulty, the mechanisms behind both these instabilities are not yet fully understood in single fuel systems, let alone multi-fuel systems. A possible solution might be found in active instability control systems that were once discarded for cheaper and less complex passive systems. In these active solutions the fuel or air flow is actively modulated to possess fluctuations that suppress the formation of combustion instabilities within the chamber.

¹for medium to high levels of enrichment

The purpose of this thesis is to advance the understanding and control of combustion instabilities in lean premixed swirl-stabilized combustors, with a focus on multi-fuel applications that include hydrogen-enriched fuel blends. Specifically, the thesis investigates the effect of forced equivalence ratio fluctuations on flame dynamics, stability, and emissions in a lean premixed swirl-stabilized combustor. By exploring how these fluctuations influence the behavior of the flame and the occurrence of instabilities, the research aims to deepen the understanding of the combustor and contribute to the development of advanced combustion control strategies. These strategies are essential for ensuring that next-generation combustors can operate safely and efficiently across a range of fuel compositions, thereby supporting the broader goals of emissions reduction and energy transition.

First, the literature study is presented in [chapter 2](#). This is followed by an explanation of the thesis outline and the combustor set-up that is used in [chapter 3](#) and [4](#), respectively. The set-points and the measuring equipment used to capture and study them are explained in [chapter 5](#). This is followed by an explanation and analysis of the experimental measurements in [chapter 6](#). Lastly, the low-order numerical analysis is presented in [chapter 7](#).

2

LITERATURE REVIEW

2.1. BACKGROUND

In this chapter a broader background for the thesis is given. This includes an overview of the major emissions of hydrocarbon (methane) combustion in [Subsec. 2.1.1](#) and how they are formed. An introduction to hydrogen and its unique properties is given in [Subsec. 2.1.2](#) as well as the possible benefits and disadvantages of using hydrogen. Additionally, the historical use and current developments with hydrogen as a fuel in both the aerospace and the industrial sectors will be discussed in [Subsec. 2.1.2](#). This includes a brief explanation of how the thesis fits in the larger APPU research project taking place at the TU Delft.

2.1.1. EMISSIONS OF COMBUSTION-POWERED SYSTEMS

Both industrial/power generation processes and aircraft engines work on the same principle of combining air and fuel, possibly premixing or atomisation and compressing the mixture, to later combust it in a combustion chamber. Through this combustion process the chemical energy of the fuel bonds is transformed into thermal energy to be used directly or to be converted into kinetic energy. For industrial processes and land based gas turbine engines, used for power generation, the most common fuel is natural gas which is mainly composed of methane. Contrary to Jet propulsion which relies on kerosene as its main fuel source. The main emission of both fuels are more or less the same and consist out of CO_2 , H_2O , NO_x , SO_2 , hydrocarbons and soot. A break down of the harmful effects of the combustion products of jet engines is presented in [Fig. 2.1](#). [\[10, 11\]](#)

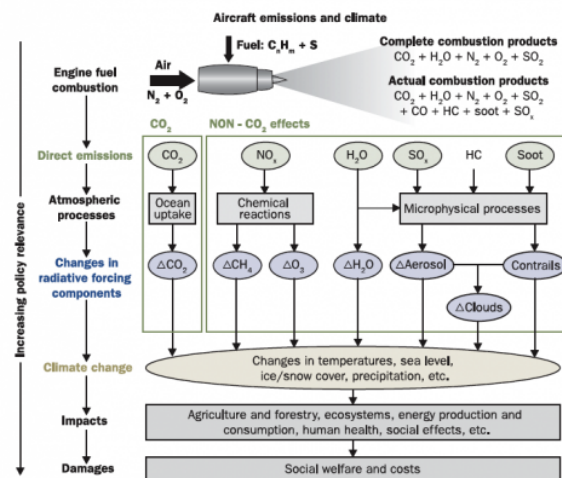


Figure 2.1: Aircraft emissions and their resulting potential impacts on climate change and welfare [\[10\]](#)

Given that the global warming potential (GWP) of (stratospheric) water is relatively low¹ compared to CO₂ emissions, research and regulations are mainly focused on the reduction of carbon emissions. Additionally, for high temperature combustion NO_x emissions, which have a significantly higher GWP, become more prevalent [14]. Therefore, the formation of both carbon and NO_x emissions will be further analysed in the subsequent subsections.

CARBON EMISSIONS

An important aspect to note when discussing emissions, particularly carbon emissions, is the role of the equivalence ratio (ϕ). The equivalence ratio represents the stoichiometric air-to-fuel ratio relative to the air-to-fuel ratio, which is the chemically balanced ratio required for complete combustion. Understanding the influence of the equivalence ratio on emissions is crucial for comprehending the combustion process.

Fig. 2.2 illustrates the influence of the equivalence ratio on various emissions, although it should be noted that the figure does not include the representation of unburned hydrocarbons (HC). Unburned hydrocarbons typically exhibit a parabolic profile skewed toward the lean side. By mentally completing the picture, one can easily clarify the trends observed.

Starting from the stoichiometric point and moving toward the fuel-rich side, an increase in carbon monoxide (CO) and HC can be observed. This increase is due to the insufficient availability of air to facilitate complete combustion of all the hydrocarbons present. On the other hand, the peak of carbon dioxide (CO₂) emissions at the stoichiometric point is logical since the stoichiometric point represents the ideal combustion condition where all the fuel is fully combusted. Intuitively, one would expect the highest temperatures to be around this point, as all the fuel is reacted away. However, in practice, the (adiabatic) temperature peak is slightly shifted toward the fuel-rich side due to additional endothermic reactions that absorb heat from the reaction on the lean side. This temperature profile is also related to the peak production of nitrogen oxides (NO_x), which will be discussed in the next section. Additionally, it is evident from the figure that as CO₂ production starts diminishing, CO production starts increasing on the fuel-rich side. This occurs due to a lack of oxygen available for complete combustion on the rich side.

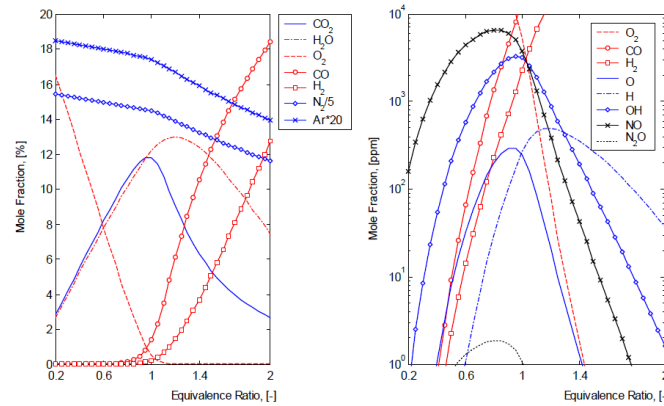


Figure 2.2: Products of equilibrium for complete combustion of kerosene (C₁₂H₂₃) combustion in air at $p = 28$ [atm], $T_{air} = 700$ [K], $T_{fuel} = 298.15$ [K] [15]

A critical point to address is the intricate nature of combustion kinetics. Multiple reversible reaction pathways exist for the formation of various products, as visualized in Fig. 2.3. These pathways elucidate the interplay between temperature, equivalence ratio, and the formation of reaction products. At lower temperatures, multiple pathways and bidirectional reactions come into play. These contribute to the formation of intermediate species, resulting in a diverse set of reaction products. As the temperature increases, some pathways become more dominant, while others may diminish in importance. This temperature-dependent behavior impacts the distribution of reaction products and their concentrations. [16]

¹The impact of contrail formation which requires the presence of water is not negligible and still being researched [12, 13].

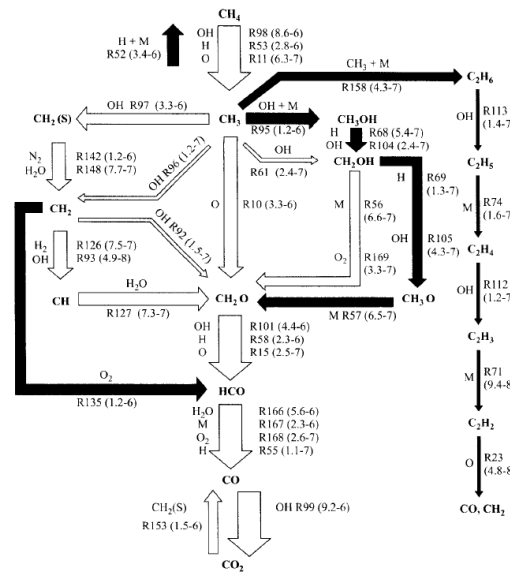


Figure 2.3: High-temperature (2200 K, white arrows) and additional low-temperature (<1500 K, black arrows) reaction pathways for CO_2 in methane combustion. [17]

A last important point to address regarding Figure 2.1 and Figure 2.3 is the concept of complete (equilibrium) combustion. The various pathways depicted in Figure 2.3 represent different reactions with varying reaction rates and timescales. It is crucial to consider the kinetics of these reactions, as some reactions, such as certain nitrogen oxide (NOx) reactions, can be quite slow. In practical combustion systems, it is essential to ensure that the combustor is appropriately sized to provide a residence time that allows for complete combustion. Residence time refers to the duration that the reactants spend inside the combustion zone or chamber. It is a combination of the length of the chamber and the flow speed of the reactants. By providing a sufficiently long residence time, the combustion process is given enough time for all the reactants to undergo the necessary reactions and achieve complete combustion. This allows for the conversion of fuel and oxidizer into the desired products and minimizes the formation of incomplete combustion byproducts or pollutants. This can be achieved through appropriate sizing of the combustion chamber or by implementing measures such as flow control, turbulence enhancement, or optimizing mixing to facilitate complete combustion.

Furthermore, the residence time also affects the extent of secondary reactions and the overall product distribution. In systems with shorter residence times, the reactants may not have sufficient time for secondary reactions, such as the conversion of nitrogen monoxide (NO) to nitrogen dioxide (NO_2) or the further oxidation of carbon monoxide (CO) to carbon dioxide (CO_2). These secondary reactions play a crucial role in achieving cleaner and more efficient combustion. However, some high temperature NOx reaction are slow in nature which might increase the overall NOx production if the residence time is increased. Hence, the phenomenon should be carefully examined during the design phase. An overview of all the NOx production pathways is given in the next section.

NITROGEN OXIDES (NOx) EMISSIONS

As alluded to in the previous section NOx production is heavily correlated to, and actually influences, the temperature of the flame as shown by Fig. 2.4. In this figure the three mechanisms that are generally considered, i.e. thermal, prompt and the NO route, are presented relative to the temperature of the flame. As one can clearly deduce the NOx production, especially due to the formation of thermal NOx, increases significantly with elevated temperatures. [5, 16–19] Another phenomena that was introduced in the previous section that is also present in Fig. 2.4 is the concept of residence time and its effect on slow reactions like (thermal) NOx production. However, findings by Leonard and Stegmaier [20] give a more nuanced image. They found no serious correlation between NOx production and residence time for thoroughly premixed flames below 1900 K in their engine that was optimised to operate at low temperature and therefore has low thermal NOx production.

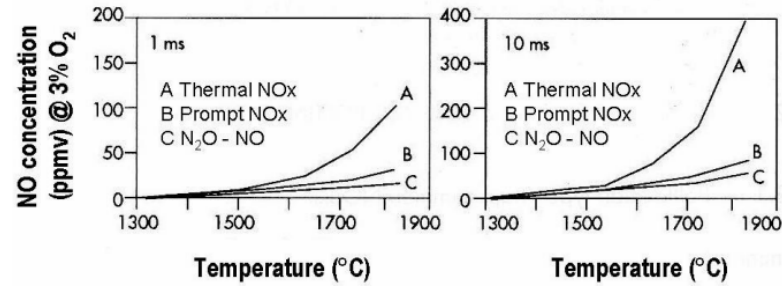


Figure 2.4: Calculated results by Zevenhoven and Kilpinen [21] showing the significance of different NO formation mechanisms as related to temperature and residence time when burning methane under stirred reactor conditions (CSTR). Pressure 1 bar, stoichiometric ratio 1.15.

Referring back to Fig. 2.2 peak NOx production sits slightly towards the lean side from stoichiometric point while the adiabatic temperature peak lies on the rich side of stoichiometric equilibrium. The reason for this can be explained using Fig. 2.5, one sees that the mechanisms for NOx formation require a lot of excess air which becomes less available the richer the mixture becomes. As most of these NOx reactions are endothermic in nature, i.e. taking energy from the environment, the temperature peak is shifted from the stoichiometric point.

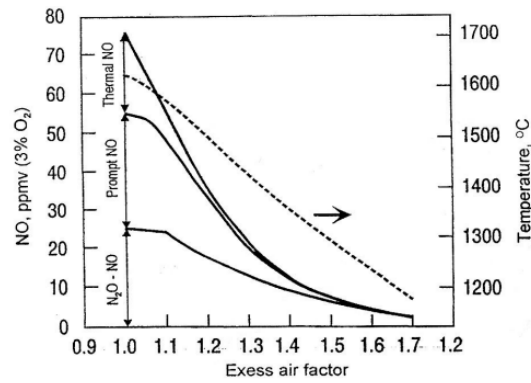


Figure 2.5: Calculated results of the significance of NO formation mechanisms versus air factor when burning methane under stirring reactor (CSTR) conditions. Pressure 1 bar, residence time 4 ms. [21]

Extended Zeldovich or Thermal NO

The dominant route for NOx at elevated temperatures is through the extended Zeldovich mechanism, originally presented by Zel'dovich [22] in 1946. The so called thermal NOx is formed by (slow) reactions between nitrogen and oxygen that require a high activation energy. This is why thermal NOx is considered to be relevant only at high temperatures above 1700K as the high temperatures allow for the dissociation of diatomic oxygen and nitrogen. High pressures will also further help fulfill the high activation energy required. [5, 16, 19, 23]

Requirements for thermal NOx:	Reactions for thermal NOx:
• Presence of N_2 • High temperature (>1700K) • High concentration of O-atom	$O + N_2 \rightleftharpoons NO + N$ $N + O_2 \rightleftharpoons NO + O$ $N + OH \rightleftharpoons NO + H$

Prompt or Fenimore mechanism

This rather fast mechanism sees the creation of NOx through the reaction of CH-radicals, i.e. C, CH, CH₂ with molecular nitrogen. [5, 16, 19, 23] The reaction happens in rich conditions, which may even occur in globally lean mixtures and does not require high activation energy meaning that it can occur at temperatures as low as 1000 K. [16] Please note that this route is only relevant for fuels with carbon containing reactants.

<i>Requirements for prompt NOx:</i>	<i>Reactions for prompt NOx:</i>
• Presence of N₂ • (Locally) rich conditions: Large CH-radical-concentration • Carbon containing fuel	$\begin{aligned} \text{N}_2 + \text{CH} &\rightleftharpoons \text{HCN} + \text{N} \\ \text{HCN} + \text{OH} &\rightleftharpoons \text{CN} + \text{H}_2\text{O} \\ \text{CN} + \text{O}_2 &\rightleftharpoons \text{NO} + \text{CO} \end{aligned}$

Nitrous Oxide route

At higher pressures, due to the need for a third body (M) in the reaction, the path via N₂O becomes important and competes with the thermal NOx to form NO. The NO forms from the intermediate N₂O molecule which results from the oxygen-attack of diatomic nitrogen.[5, 16, 23] The need for the third body to part-take in the reaction lowers the activation energy required for this reaction making the route less susceptible to lower temperatures than thermal NOx. Given the high pressure this route is especially relevant for lean gas turbine engines as the route is promoted in lean combustion. [23–25]

<i>Requirements for nitrous oxide route NOx:</i>	<i>Reactions for nitrous oxide route NOx:</i>
• Presence of N₂ • High concentration of O-atom • High pressure	$\begin{aligned} \text{N}_2 + \text{O} &\rightleftharpoons \text{N}_2\text{O} \\ \text{N}_2\text{O} + \text{O} &\rightleftharpoons \text{NO} + \text{NO} \\ \text{N}_2\text{O} + \text{H} &\rightleftharpoons \text{NO} + \text{NH} \end{aligned}$

NNH Route

An additional pathway of NOx formation was presented by [Bozzelli and Dean](#) [26] after its potential kinetic significance was suggested by [Miller et al.](#) [27]. The pathway is relevant for the entire range of equivalence ratios, especially at low residence times as the mechanisms occurs in the flame brush and other areas where high concentrations of H and O radicals are present. [23, 24] There is experimental evidence [28] for the NNH mechanism occurring in low-temperature, low-pressure, premixed rich flames of hydrogen and air. The mechanism is also a major pathway for rich mixtures in combination with low residence times. At stoichiometric condition with temperatures of exceeding 2100 K the thermal NO route is said to become dominant after only for residence times larger than 1 ms. [23]

<i>Requirements for NNH NOx:</i>	<i>Reactions for NNH NOx:</i>
• Presence of N₂ • (Locally) Rich conditions • Large O- and H-radical-concentration	$\begin{aligned} \text{N}_2 + \text{H} &\rightleftharpoons \text{NO} + \text{NNH} \\ \text{NNH} + \text{O} &\rightleftharpoons \text{NO} + \text{NH} \end{aligned}$

Fuel NOx

Generally the conversion of fuel-nitrogen or fuel NOx, depicted in [Fig. 2.6](#) is also identified as a key mechanism. The mechanism is especially relevant for nitrogen containing fuels like coal and produces a considerable amount of NOx irrespective of flame temperature.[16, 21] However, for methane fuel the process is not relevant.

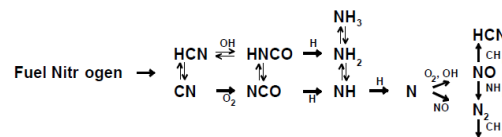


Figure 2.6: NO-formation pathways through the fuel-nitrogen mechanism[16]

N₂H₃ Route

A sixth route for NOx production was proposed by [Konnov and Ruyck](#) [29], who state that it is possible that NO can be formed through N₂H₃. The complete pathway from NH₃ to NO is presented as: NH₃ → NH₂ → NH → N → NO. However, the mechanism is stated to be relevant for rich mixture at low temperatures (<1500 K). [19, 23]

2.1.2. HYDROGEN AS A FUEL

Throughout history, especially during times marked by fossil fuel crises and environmental challenges, hydrogen has consistently been heralded as a promising alternative to conventional fossil fuels. This recognition is largely due to its potential for clean production from water through electrolysis, coupled with its status as the most abundant element in the universe. The carbon-free emissions resulting from hydrogen combustion have sparked significant interest and attention, underscoring its viability as an eco-friendly energy source [11, 30]. In recent years, the shift towards hydrogen as an established energy source has become more viable and practical than ever before. It is increasingly recognized as an essential solution capable of addressing the urgent need to reduce dependence on traditional energy sources, such as fossil fuels, while complementing renewable energy sources like solar and wind power [31, 32]. Through electrolysis using surplus electrical power from renewables, hydrogen can be cheaply generated and used for a variety of purposes like combustion, transport, chemical feedstock and long term energy storage. [30, 33]

In the context of increasingly stringent emission regulations due to the rising concern over global warming, hydrogen emerges as a viable solution for combustion processes. Fig. 2.7 illustrates how hydrogen, devoid of carbon and sulfur compounds, substantially reduces or eliminates CO_2 emissions[11]. This approach primarily leaves NO_x emissions as the primary environmental concern.

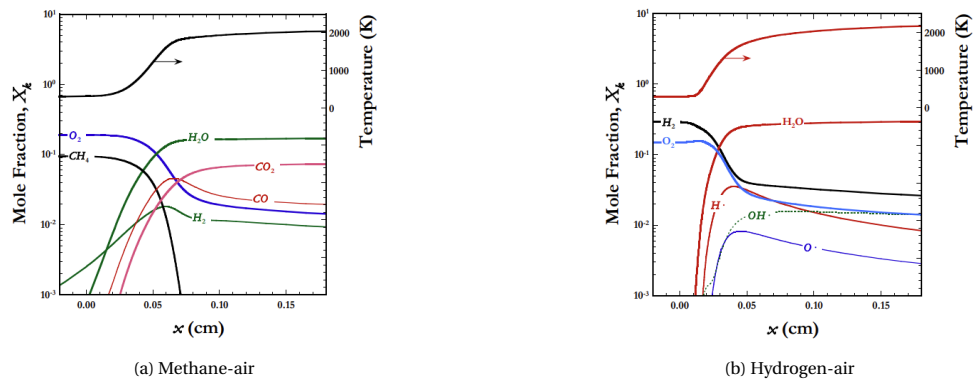


Figure 2.7: Flame structures of stoichiometric flames at the ambient pressure, computed using the GRI-Mech. [34]

First a brief overview of how hydrogen has been used in both the aviation and industrial sectors thus far will be presented in Subsec. 2.1.2.1. Then the general properties will be discussed in Subsec. 2.1.2.2 followed by the advantages of hydrogen as a fuel for combustion in Subsec. 2.1.2.3. Lastly, the various technical challenges associated with hydrogen for combustion in Subsec. 2.1.2.4.

HISTORICAL USE AND CURRENT DEVELOPMENTS OF HYDROGEN AS A FUEL

This section details the historical use and current use of hydrogen in both the aviation and industrial sectors. It also provides a brief outlook of current developments in both sectors as well as an in depth presentation of the APPU project which is the TU Delft's Advanced Propulsion & Power Unit initiative aimed at revolutionizing aircraft propulsion systems for greener and more sustainable aviation. This project forms the overarching frame work for this thesis.

Aviation

The first jet engine that ran on hydrogen, Heinkel-Strahltriebwerk 1 (HeS 1), was already created back in 1937 by Von Ohain and Ernst Heinkel. The first hydrogen-powered flight took place 20 years later, in 1957 with a B-57B bomber. The test was conducted by NASA and used a modified jet engine with liquid hydrogen next to a normal kerosene system. The next hydrogen-powered plane only arrived in 1988 by the soviets, who transformed a TU-154 into a cryogenic test bed for liquid hydrogen. This was followed in 1991 by the hydrogen powered A310 project by airbus which would form the basis for the liquid hydrogen Cryoplane project in 2000. Recently airbus announced that it would build its ZEROe demonstrator on a A380, to allow for various novel systems, including hydrogen jet engines to be tested. [35–38]

It should be noted that all these projects saw the modification of the jet engines to allow for pure hydrogen combustion. The converted engines could not use a different fuel or fuel blend. A single engine that can

run on both hydrogen and kerosene still needs to be developed. Meanwhile, the urgency for transitioning to more sustainable fuels has grown over the past years. Whereas, the lack of a worldwide (green) hydrogen infrastructure has been persevered. Hence, it has become evident that hydrogen enrichment of existing fuels and overall fuel flexibility should be the initial focus of research instead of complete system overhauls. This is shown in an increase in academic research regarding the topic in the last years.

Next to hydrogen for combustion some parties also look into incorporating hydrogen fuel cells instead for smaller short range aircraft. The fuel cell which uses the reverse electrolysis reaction to form water and electricity. [37, 39] Apart from hydrogen biofuels, electrofuels and even fully electric systems are considered for sustainable aviation, a recent comparative overview was written by Su-ungkavatin *et al.* [40].

APPU: Advanced Propulsion & Power Unit

The Auxiliary Propulsive and Power Unit, or APPU in short, project [8] is funded by the dutch government (PPS grant) and industry. The project is a collaborative effort by coordinated by Dr. Gangoli Rao and Prof. Leo Veldhuis from the TU Delft, in partnership with SAFRAN group, Airbus, and Rotterdam the Hague Innovation Airport (RHIA). The proposed APPU system is a novel system that replaces the traditional APU with a multi-functional state of the art gas turbine in a boundary layer ingestion (BLI) configuration improving the overall aircraft efficiency. The system should be able to operate on both Hydrogen and/or kerosene fuel, depending on local availability. It can also drive a variable pitch open rotor propulsion system at the aft end of the fuselage increasing the usage of the APPU.



Figure 2.8: The APPU project goals and partners. [41]

Project Goals:

- Design a BLI system for A320 type of Aircraft
- Design and test an open rotor propulsion system operating in boundary layer ingestion regime
- Design and test a hydrogen combustor for the APPU engine
- Design and optimize the aircraft empennage and performance for next generation A320 with the APPU system
- Investigate storage solutions for H2 in an aircraft

One of the main technological challenges that is addressed is the development of a hybrid combustor that is capable of handling both hydrogen and kerosene. Several different combustors have been proposed for the purpose of pure hydrogen combustion, such as the micro-mixing concept [42, 43], lean direct injection [44, 45] and swirl stabilized with axial (air) injection [5, 46]. However, none of these have been adapted to accommodate multiple fuels or fuel blends. For the APPU project it has been decided to further develop the

swirl stabilized combustor with axial (air) injection to allow for both kerosene and hydrogen combustion. To this end a swirl stabilized with axial (air) injection has been designed and tested with methane at the TU Delft sustainable aircraft propulsion lab. [47] This thesis will experimentally research the hydrogen enrichment of methane, which is a simpler (to handle) and more widely researched molecule than kerosene.

The other novel technology aspect of the project is the aforementioned BLI. This technique requires substantial changes to the aircraft's structure and the incorporation of its engines but if done correctly it can offer several benefits. [48–50] However, as this thesis focuses on the hydrogen combustion part the aspect of BLI will not be further discussed.

Industrial Applications

Hydrogen finds diverse applications in the industrial sector, serving as a vital chemical feedstock, facilitating oil refinery processes, and supporting iron and steel production. At present, a significant portion of hydrogen is utilized in ammonia production (NH_3) for fertilizer manufacturing and hydrocracking operations in oil refinement. Nonetheless, the potential for hydrogen's application as a high-temperature fuel replacement for natural gas or as a co-firing agent with natural gas is promising, particularly with the anticipated decline in hydrogen prices [6]. Despite its promise, the adoption of hydrogen as a fuel source for industrial high-temperature applications has been relatively limited, primarily restricted to academic studies [51]. Large-scale combustion production processes or real-world pilot projects remain scarce. However, this landscape is expected to transform, driven by the increasing availability of cost-effective renewable energy sources for hydrogen production and the governments' strong support through subsidies and regulations [6].

A benefit of industrial gas turbines and other combustion facilities, compared to aircraft engines, is that these land-based systems allow for a wider exhaust gas treatment possibilities as they are less limited by weight and space constraints. For example water can be injected at certain points or the exhaust gases can even be captured and cleaned by a selective catalytic reduction. This makes issues such as higher NO_x emissions due to higher flame temperature more straight forward to deal with. [52]

A challenge associated with using hydrogen for heat generation and high-temperature applications, that is not applicable for aero-engines, lies in its comparatively lower radiative heat transfer capacity compared to conventional fuels [53]. Addressing this concern, one viable solution involves introducing other media, such as clinker dust, into the fuel stream, effectively augmenting the radiative heat transfer coefficient of the hot combustion gases [6].

GENERAL PROPERTIES OF HYDROGEN

A key characteristic of hydrogen is its high adiabatic flame temperature (T_{ad}) as shown in Fig. 2.9. The adiabatic flame temperature is defined as "The temperature achieved by the flame if the net energy liberated by the chemical reaction was fully utilized in heating the combustion products." [5]. In practice it rarely occurs due to heat losses through convection and radiation. Regardless it can still be used as a good indicator for the to be expected flame temperatures.

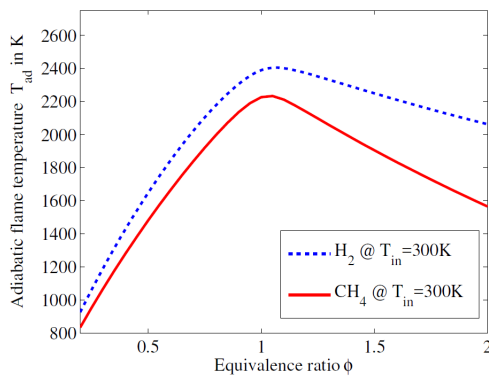


Figure 2.9: Calculated adiabatic flame temperature for methane and hydrogen by Reichel [5]

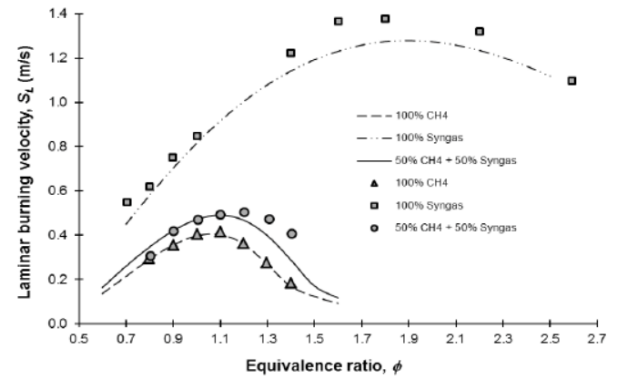


Figure 2.10: Laminar burning velocities for methane, syngas and hydrogen adapted from Cardona *et al.* [54]

Other relevant properties of hydrogen relative to methane regarding combustion are presented in [Table 2.1](#) and explained below.

Table 2.1: Several relevant characteristics of hydrogen compared to methane [5, 30]

Property	Unit	Hydrogen	Methane
Density	kg/m^3	0.084	0.65
Lower Calorific Heating Value (LHV)	kWh/kg	33.33	13.9
Lower Calorific Heating Value (LHV)	$GJ/mole$	2.412	0.802
Stoichiometric volume fraction in air	vol %	29.53	9.43
Diffusion coefficient in air	cm^2/s	0.61	0.16
Specific heat at constant pressure	Jg/k	14.89	2.22
Flammability limits in air	vol %	4.0-75.0	5.3-15.0
Ignition energy in air	mJ	0.02	0.29
Ignition temperature	K	858	813
Flame emissivity	%	17-25	25-33
Stoichiometric Laminar flame speed (S_L)	cm/s	210	40
Quenching distance (δ_q)	mm	0.64	2.5

Hydrogen fuel is also known for having the highest gravimetric energy density of any known substance as shown in [Fig. 2.11](#). However, this is offset by its very low volumetric density. This means that hydrogen needs little mass but a large volume to store its energy. This explains both the high LHV and volumetric fraction needed in air presented in [2.1](#).

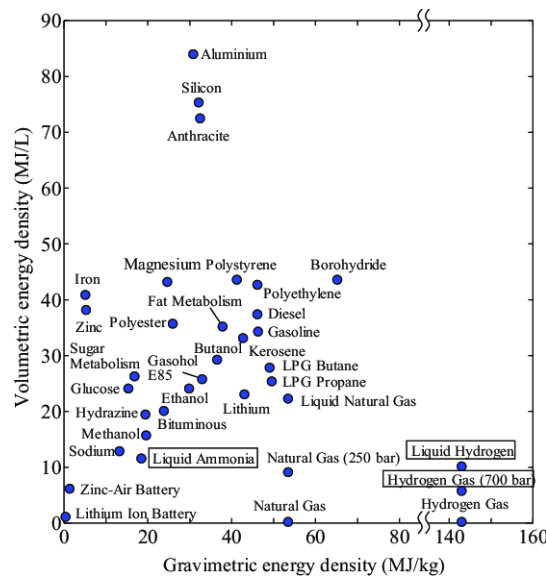


Figure 2.11: Volumetric and gravimetric energy density graph for a variety of fuels by [Kobayashi et al. \[55\]](#)

Hydrogen also diffuses very quickly as indicated by the high diffusivity coefficient in air, this means that the mixing properties of hydrogen are far better than methane. Its high specific heat, which indicates how much energy is necessary to increase its temperature, can be attributed to the small size of the hydrogen molecule. Both of these aforementioned parameters explain the wider flammability range, low activation or ignition energy and higher ignition temperature. Given its high diffusivity ignition spreads easily throughout the mixture igniting the entire flow if the temperature is sufficiently high. Its high activation temperature, which can be attributed to its smaller size which causes less collisions compared to bigger molecules, goes hand in hand with its high specific heat capacity as the lack of collisions at lower temperatures make it harder to increase the temperature and to generate enough successful collisions that start the combustion reaction. Furthermore, the same reasoning can be used to explain the higher (laminar) flame speed and small quenching distance. In simplified visualisation, as shown in [Fig. 2.12](#), a flame should be seen as region which is high in reactants

(fuel and oxygen possibly mixed) and a region with products which are separated by a flame front. This flame front, wherein the reaction takes place, travels into the reactants with a certain speed and can be countered by giving the input flow the same speed to create a stationary flame. If the flame speed is higher than the incoming flow one risks flashbacks, vice versa one risks blow out. The flame front consists out of a preheating zone, where the mixture ignites, a reaction zone where the main reaction takes place and a post-combustion zone where the products continue to react in secondary reactions. The flame speed indicates how fast the active flame front, where the oxidation reaction takes place, moves into to reactant flow.

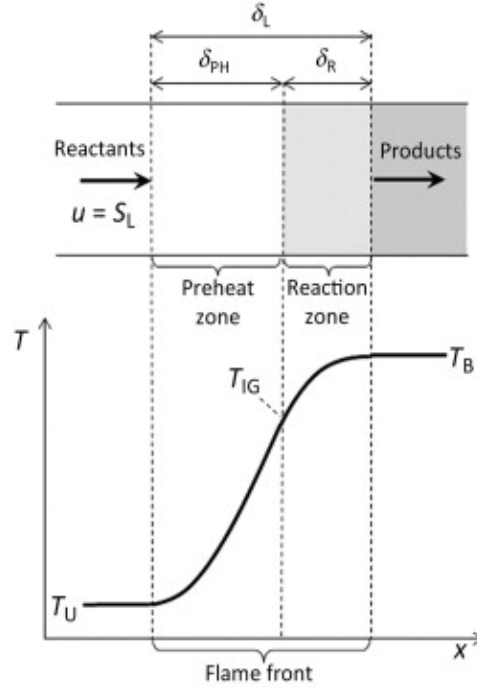


Figure 2.12: Illustration of a one-dimensional planar and unstretched flame front. [56]

Lewis Number

One fundamental way hydrogen differs from conventional hydrocarbon fuels is captured by the Lewis number. This dimensionless number relates the thermal diffusivity of a substance to its mass diffusivity as seen in Eq. 2.1. It basically tells one how the temperature gradient in a fluid compares to the concentrations gradient. In case the Lewis number is much lower than 1 the mass transfer dominates over the heat transfer. Hydrogen, which possesses a Lewis number of 0.33, behaves inherently different from the hydrocarbon fuels which have a Lewis number greater than 1. This means that for hydrocarbons, heat transfer often dominates, while for hydrogen, mass transfer tends to be more critical, resulting in different combustion characteristics and flame behaviors.

$$Le = \frac{\alpha}{D_{AB}} \quad (2.1)$$

BENEFITS OF USING HYDROGEN

As already alluded to in the section's introduction the main benefit of using hydrogen as a fuel is the complete lack, or strong decrease in case of enrichment of a fossil fuel, of carbon atoms. Causing drastic reductions in CO₂ and its associated minor species (CO, hydrocarbons and soot) as well as SO₂ emissions. The effect is already noticeable when it is blended with other fuels as shown in Fig. 2.13. It should be noticed that when the same graph is made based on energetic blend ratio it linearises as hydrogen has a higher heating value. In case liquefied hydrogen is compared to kerosene as an aircraft fuel, the global warming potential of the fuel drops from 1.05 kgCO₂eq/tonne-km, for kerosene, to 0.015–0.19 kgCO₂eq/tonne-km² for liquid hydrogen according to the life cycle analysis of aircraft fuels conducted by Bicer and Dincer [57].

²Depending if it is made via solar power or geothermal

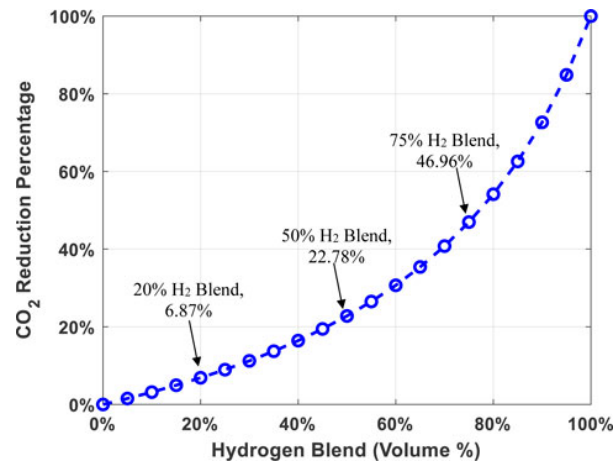


Figure 2.13: CO₂ Reduction with respect to hydrogen and methane blend percentages [58]

A second benefit of hydrogen can be attributed to its high flammability, not only does it have a low ignition energy but high flame speed and diffusion properties. Although some of these properties come with strong drawbacks, they do mean that hydrogen possesses a wider flammability range relative to the equivalence ratios of the possible mixtures as shown in Fig. 2.14. Generally, Hydrogen mixture ignite easily and the flame spreads quickly. An additional benefit of this wider range is that it permits one to burn a lot leaner compared to other fuels, further reducing emissions. However, this trade-off once again comes with some remarks which will be discussed in the next section.

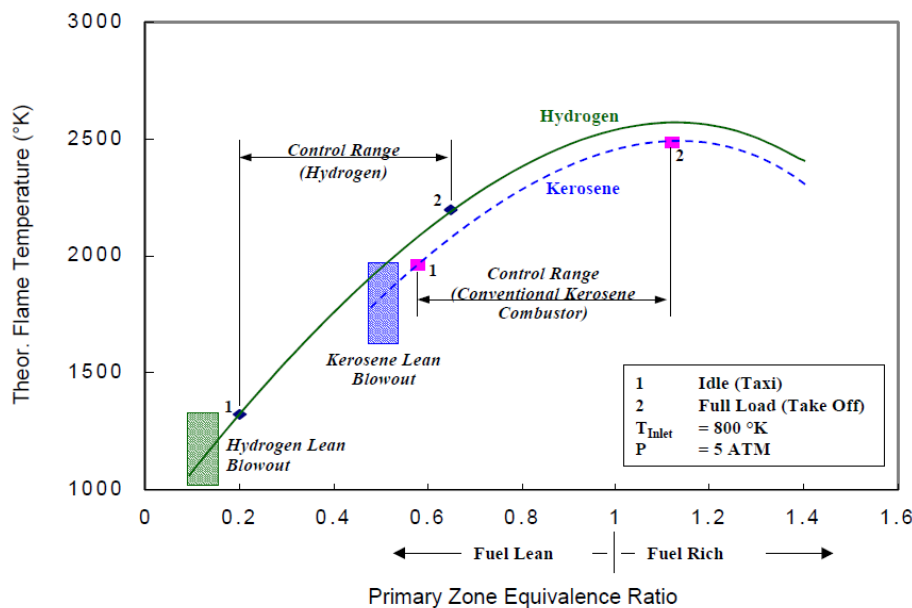


Figure 2.14: Equivalence ratio control range for hydrogen and kerosene combustion [59]

DISADVANTAGE(S) OF USING HYDROGEN

Hydrogen poses a large variety of technical challenges and questions associated to its unique properties in the various aspects for its use. These range from broader questions regarding optimum storage and transportation methods[35, 36] to technical aspects like hydrogen embrittlement of metals[60] caused by the high diffusivity, leak detection, ect.. Similarly, there exist a subset of technical challenges specifically associated to the nature and intricacies of hydrogen combustion. Even though, aspects such as material compatibility and storage method play an important role for the eventual success of hydrogen as an (aviation) fuel, the scale of the problem(s) in combination with the fundamental combustion focus of this research has lead to the decision to limit the scope in this study to the fundamental combustion aspects.

In this light there is one major drawback associated to using hydrogen from an application point of view with regards to flame anchoring and flashback that can be attributed to the higher flame speed, diffusivity and tendency to auto-ignite. Flashback consists of an uncontrolled phenomena, common in premixed combustors, where the flame travels upstream into the injector and mixing tube and causes great damage. Its possible causes differ but all entail local imbalances in the flow with regards to flame speed, which is associated to gas composition, and flow velocity. The various causes will be covered in depth together with their solutions in the next chapter, [Sec. 2.2](#), after an introduction of lean, turbulent, (swirling) combustion.

Also from an emissions point of view. The higher adiabatic flame temperature associated with hydrogen established in [Subsec. 2.1.2.2](#), feeds an exponential rise in thermal NO_x formation as seen in [Subsec. 2.1.1.2](#). Thus for equal equivalence ratios hydrogen will generate more NO_x than kerosene or methane. As a matter of fact, the previously mentioned low ignition energy and high diffusivity of hydrogen, allow for a much broader equivalence operating range as shown in [Fig. 2.14](#). This broader range allows one to burn much leaner and cooler compared to kerosene or methane which reduces thermal NO_x formation, which is the dominant pathway for high temperature applications. [5] Ultra-lean hydrogen systems are currently being researched for equivalence ratios below 0.4. [7, 61]

However, one should realise that eventually, by lowering the adiabatic flame temperature one is in fact decreasing the turbine inlet temperature and thus the power³. Furthermore, many applications in aviation and industry rely on principles of combustion to achieve the necessary elevated temperatures and energy outputs.

2.2. FUNDAMENTAL BACKGROUND

In this chapter the necessary background regarding combustion in a swirl stabilised combustor is presented. First a brief explanation of the relevant combustion conditions is provided in [Subsec. 2.2.1](#). This is followed by a theoretical section on turbulent premixed combustion in [Subsec. 2.2.2](#).

Then typical concerns for stable combustor design will be covered. An overview of the most relevant parameters is provided in [Fig. 2.15](#). One sees clear limits that define the stable combustion region. These limits are also affected by several parameters such as fuel composition, operating pressure and temperature. As alluded to in the last chapter, in case of hydrogen enrichment the flashback boundary becomes increasingly more important with increasing hydrogen content. Therefore, the topic is discussed in [Subsec. 2.2.3](#). The remaining limitations like extinction, blow-out and blow-off will be briefly discussed in [Subsec. 2.2.4](#). Combustion instabilities, like thermoacoustic instabilities, will be covered in [Sec. 2.5](#).

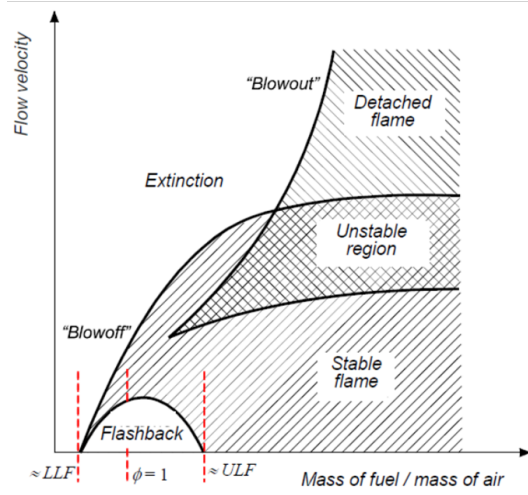


Figure 2.15: Typical flame stability diagram including the most relevant regions and the Lower and Upper Limit Flashback (LLF and ULF) [62]

The next section, [Subsec. 2.2.5](#), is dedicated to detailing the principles and dynamics of swirl-stabilised combustors. It also covers relevant mechanism such as swirl decay and the effects on combustion.

³If the temperature becomes lower than the original with methane for the same mass flow

Lastly, Axial Air Injection, a more novel method that is used to decrease the flashback propensity of a swirl stabilised combustor, is introduced in [Subsec. 2.2.6](#) given that it is used in the set-up at the TU Delft sustainable aircraft propulsion lab

2.2.1. RELEVANT COMBUSTION CONDITIONS: LEAN PREMIXED TURBULENT SWIRL-STABILIZED COMBUSTION

Generally combustion and flames are classified among three major categories, lean or rich, premixed or non-premixed and laminar or turbulent, identifying the core characteristics of the flame.

The definition of a rich or lean flame is related to the equivalence ratio (ϕ) and was already introduced in the previous chapter. Naturally, in very turbulent or not perfectly premixed combustion the equivalence ratio varies throughout the flame, resulting in these richer and, thus, hotter pockets in the lean flame or vice versa. When analysing NOx emissions it is critical to analyse these hot pockets as here the bulk of NOx emissions is generated.

Premixing is a common technique used to create a more uniform mixture such that the wanted equivalence ratio is achieved throughout the mixture in order to have more control over emissions. It is accomplished by mixing the air and fuel in a premixing tube before dumping it in the combustion chamber where it gets ignited. This is completely opposite of diffusion or non-premixed flames where the air and fuel streams are injected separately into the combustion chamber and the flame front is created where these two meet within the right concentrations, typically this is at stoichiometric conditions. Naturally, this will create more NOx and

The difference between laminar and turbulent combustion can be explained by looking at the aerodynamic laminar and turbulent flow. Similarly, Laminar combustion produces a smooth flow with a single well-ordered flame front. Turbulent combustion is much more chaotic in nature and possesses complex flame structures that will be covered in [Subsec. 2.2.2](#).

The swirl-stabilisation refers to the flame stabilisation technique that is used to promote flame anchoring and ensure proper mixing of the oxidiser and fuel. A swirl is given to the flow, via swirling vanes or by tangential injection, resulting in a swirling central jet as shown in [Fig. 2.16](#). This central jet will then decay and form a recirculation zone in which the flame can anchor. The concept will be further clarified in the subsequent section.

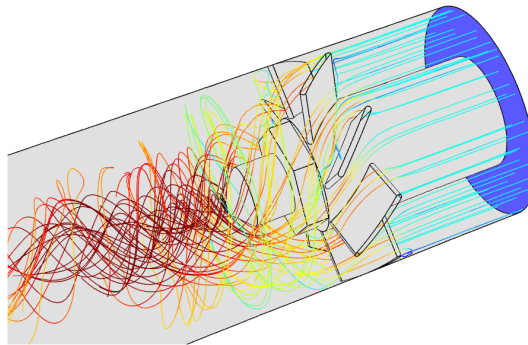


Figure 2.16: Streamlines generated by axial swirler visualised through numerical simulation (ANSYS) [63]

Next to swirl stabilisation, micromix diffusive combustion and bluff body flame holders have also been used for (experimental) hydrogen combustor designs. The former entails creating a blanket of small flames by injecting the air axially while the hydrogen is added perpendicularly via small holes. This mixture is then combusted to form the various little flamelets which will create an inner and outer recirculation zone. The latter sees the placement of a bluff body at the exit of the injector where the disturbance it generates will anchor the flame and create recirculation zones. However, both of these techniques use mechanical stabilisation whereas swirl-stabilized flames depend on aerodynamic stabilisation. The aerodynamic stabilisation reduces the contact between the flame and the inlet tube which reduces the heat load on them and hence prolongs combustor life time. [5]

2.2.2. TURBULENT PREMIXED COMBUSTION

The more fundamental aspects of turbulent premixed combustion will be covered in this section, like the different modes that can be distinguished and how the turbulent flame speed which differs from the laminar can be determined.

TURBULENT PREMIXED COMBUSTION THEORETICAL MODES

The different flame types are categorised by the intensity of the turbulence and its ability to affect the flame structure itself. The turbulent velocity fluctuations are able to deform and breakdown depending on the characteristics the eddies in the flow possess. Moreover the various turbulent premixed flames can be classified through the scale and intensity of the flame thickness, the laminar burning velocity of the mixture and the speed and magnitude of the turbulent flow eddies. How the time scale of these eddies relate to the chemical time scales of the turbulent combustion reaction is captured in two dimensionless numbers: the Karlovitz and Damköhler number. These help in the assessment whether or not a laminar flame structure could survive in the turbulent flow.

Turbulent Karlovitz Number

The turbulent Karlovitz number (Ka) relates the timescale of the chemical reaction (τ_c), which is equal to that of a laminar flame of the mixture, to the Kolmogorov timescale (τ_η), which is that of the smallest turbulent eddies in the flow, as shown in Eq. 2.2.

$$Ka = \frac{\tau_c}{\tau_\eta} \quad (2.2)$$

The chemical time scale (τ_c) is given by Eq. 2.3, is defined by the laminar flame speed (S_L) and the thermal diffusivity (α) which can be related to the flame thickness ($\delta_f = \alpha/S_L$).

$$\tau_c = \frac{\alpha}{S_L} \quad (2.3)$$

The Kolmogorov scale is the level of the smallest eddies in the flow. The time it takes for these eddies to rotate is equal to the time for them to diffuse. The Kolmogorov timescale (τ_η) which is a key characteristic of the Kolmogorov scale is defined by Eq. 2.4. Where η defines the characteristic length of the eddies and ν stands for the kinematic viscosity. It is important to note that the flow can be considered laminar for dimensions below this characteristic length scale.

$$\tau_\eta = \frac{\eta^2}{\nu} \quad (2.4)$$

When the Kolmogorov scale is greater than the flame thickness or chemical scale one can define the system as a locally laminar premixed flame embedded in a turbulent flow. This region where $Ka < 1$ is where the flamelet regime holds and is illustrated in the Borghi diagram, Fig. 2.17, below the $Ka = 1$ line. Under this regime a clear and distinct flame front can be seen in the flame. However, due to the increase in surface area and thus contact with the unreacted flow and the improved transport qualities due to turbulence the flame speed will already be significantly higher with respect to the laminar burning speed. [16]

Turbulent Damköhler Number

The turbulent Damköhler number on the other hand relates the macroscopic timescale (τ_0), i.e. the timescale of the largest eddies, to the chemical timescale (τ_c) as shown in Eq. 2.5. For example, when time needed for chemical change is greater than that of the largest eddies the eddies will be completely embedded in the reaction zone causing the notion of a distinct flame front to disappear in this well-stirred reaction. The macroscopic or integral timescales is similar to the Kolmogorov scale but instead of being that of the smallest eddies it defines the diffusion time of the largest eddies. The breakdown of these large eddies will in fact eventually feed the Kolmogorov eddies through the energy cascade.

$$Da = \frac{\tau_0}{\tau_c} \quad (2.5)$$

The regime for which the $Da < 1$ can be regarded as the well-mixed or perfectly stirred regime. For the region in between this regime and the earlier mentioned flamelet regime the distributed reaction regime sits. Where the flamelet reaction does not embed any scale eddies, the distributed only embeds a fraction and the well-mixed entails all of them. [16]

The Borghi Diagram and the Different Flame Modes

The Borghi-Peters diagram, shown in Fig. 2.17, classifies and illustrates the various region based on the Ka and Da . The horizontal axis represents the ratio of the aforementioned integral turbulent length scale and the flame thickness. The vertical axis depicts the velocity fluctuations relative to the laminar flame speed.

Naturally, if the turbulent Reynolds number, which relates a flows internal forces to the viscous forces, is not exceeded the flame will remain laminar. It can be calculated using Eq. 2.6.

$$Re_t = \frac{l_0 u'_0}{S_L \delta_f} \quad (2.6)$$

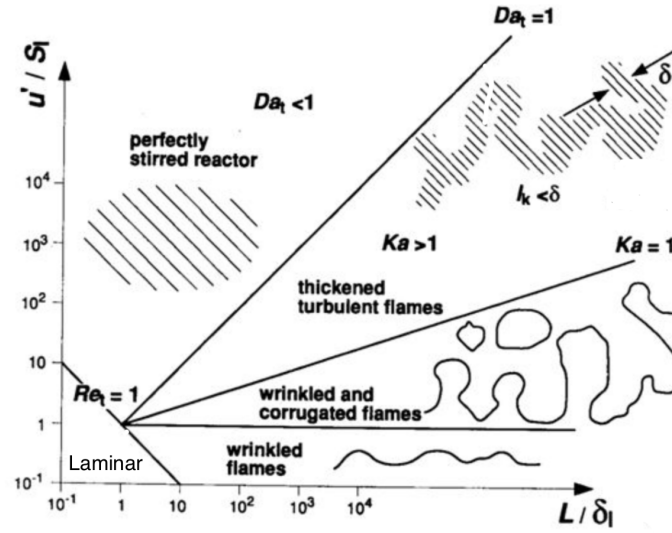


Figure 2.17: Borghi-Peters diagram depicting the various flame modes in turbulent premixed flames modified from [64]

Analysing the diagram one can notice that longer the chemical time scale is, i.e. the combustion reaction time, the more the turbulent eddies of all scales will breakdown and diffuse the distinct flame front seen in laminar flames. The various modes are elucidated below:[16]

Laminar flame: $Re_t < 1$

The turbulence if even occurring has no effect on the flame front at all and hence the flame front is simply laminar.

Wrinkled flamelets: $u'_0 < S_L$ and $Ka < 1$ and $Da > 1$

The turbulence starts to deform and wrinkle the flame front, however one can still speak of a laminar flamelet that is embedded in the turbulent flow. Given that the velocity of the fluctuations is still smaller than the flame speed the wrinkling is limited.

Corrugated flame: $u'_0 > S_L$ and $Ka < 1$ and $Da > 1$

The fluctuations speed is now higher than the flame speed causing the flame front to heavily deform, small islands away from the flame front form. The convoluted flame contains pockets of unburned mixture.

Thickened turbulent flames: $u'_0 > S_L$ and $Ka > 1$ and $Da > 1$

The flame front is widened and even torn in some places as the smallest eddies are embedded in it and affect the flame front in the preheating zone where the additional turbulent mixing promotes heat and mass transfer. This thin reaction sheet is formed by both diffusion as turbulent mixing.

Perfectly stirred reactor: $u'_0 > S_L$ and $Ka > 1$ and $Da < 1$

Almost all of the turbulent eddies affect the flame front and create a broken reaction in which no distinct local flame structure can be distinguished. The smallest eddies can enter into the reactive-diffusive flame structure, however they are still larger than the inner layer thickness.

Broken/distributed reactor: (not indicated in the figure) $u'_0 > S_L$ and $Ka > 100$ and $Da < 1$

The smallest eddies are able to penetrate into the inner layer and perturb it such that chemistry breaks down locally due to enhanced transport of heat to the preheating zone with a resulting loss in radicals and temperature decrease. This will extinguish the flame locally and fuel and oxidizer will interdiffuse and mix at lower temperatures where combustion reactions have ceased. This regime usually avoided for premixed systems as the mixing is faster than the chemistry. [65]

An important notion with this graph is the fact that in turbulent flows there is a wide range of dissipation (ϵ). Dissipation is the loss of turbulent kinetic energy due to viscous effect in the flow. Given that this is not constant throughout the flow so are the integral and Kolmogorov characteristic timescales and lengths. Therefore, a turbulent premixed flame is not represented by a single point on this graph but rather by a region that may stretch over several flame modes. [16]

TURBULENT FLAME SPEED

As already alluded to previously the turbulent flame will possess a greater burning velocity relative to its laminar counterpart. Reasons can be found in the increase contact area due to wrinkling of the flame front, improved mixing and enhanced transport properties. The first theoretical expressions for the turbulent burning velocity were formulated by Damköhler [66] who also established the earlier presented Damköhler number. In line with the role the Da plays in distinguishing flame types he identified two different turbulent burning velocity regimes, namely small and large scale turbulence, for premixed turbulent combustion. The small scale regime is also referred to as corrugated flamelets regime and the large scale as the thin reaction zones regime. He started from the conservation of mass, Eq. 2.7 in the turbulent flame surface shown in Fig. 2.18. [67]

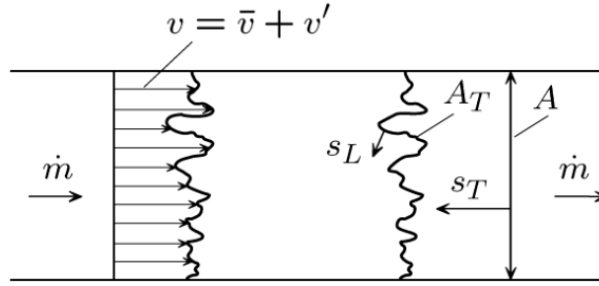


Figure 2.18: Schematic illustration of turbulent flame front [16]

$$\dot{m} = \rho_u s_L A_T = \bar{\rho}_u s_T A \quad (2.7)$$

Assuming a constant density ρ_u and averaging the flame surface area at any length scale within the inertial range one can establish Eq. 2.8.

$$\frac{s_T}{s_L} = \frac{A_T}{A} \quad (2.8)$$

Large Scale Turbulence

When turbulence scales are larger than flame thickness, turbulence increases surface area and the interaction between the turbulent flame front and flow field can be considered purely kinematic. Damköhler used the geometrical analogy with a Bunsen flame to relate the increase of the wrinkled flame surface area to the velocity fluctuations divided by the laminar burning velocity, as shown in Eq. 2.9. [67]

$$\frac{A_T}{A} \sim \frac{v'}{s_L} \quad (2.9)$$

Combining Eq. 2.9 with the earlier derived Eq. 2.8 leads to Eq. 2.10 in the limit of large v'/s_L which is a kinematic scaling.

$$s_T \sim v' \quad (2.10)$$

For weak turbulence on the other hand, simple geometric arguments [Eq. 2.11](#) can be established. [\[68\]](#)

$$\frac{s_T}{s_L} = \left(1 + \left(\frac{v'}{s_L} \right)^2 \right)^{1/2} \quad (2.11)$$

Small Scale Turbulence

When turbulence scales are smaller than flame thickness, turbulence modifies the transport process. This allows one to relate the laminar and turbulent burning speeds to the respective molecular diffusivity, D , and the turbulent diffusivity, D_T , while the chemical time scale remains the same meaning it is not affected by turbulence. Both diffusivities can be rewritten as a product of their respective velocity scales and the length scales. Meaning that for small turbulence the burning velocity ratio is not only affected by the velocity ratio v'/s_L but also by the length scale ratio ℓ/ℓ_F .

$$\frac{s_T}{s_L} \sim \left(\frac{D_T}{D} \right)^{1/2} = \left(\frac{v'}{s_L} \frac{\ell}{\ell_F} \right)^{1/2} \quad (2.12)$$

The assumption regarding the chemical time scale not being affected breaks down when the eddies penetrate into the thin reaction zone. Therefore there is an upper limit for [Eq. 2.12](#) and the small scales regime at $Ka = 1$. [\[67\]](#)

Unified Relation

There were many attempts to combine both expressions, [Eq. 2.10](#) and [2.12](#), formulated by [Damköhler](#) into a single expression that is able to reproduce the various experimental findings with regard to turbulent burning velocities. This is achieved by introducing a constant C , which is dependant on length scale ratio ℓ/ℓ_f , and an adjustable exponent n that varies between $0.5 < n < 1$ as indicated in [Eq. 2.13](#). This relation includes the limit where the turbulent flame speed becomes equal to the laminar in case that the $v' \rightarrow 0$. The exponent n is often found to be in the vicinity of 0.7 in experimental data, however attempts to justify a single exponent on the basis of dimensional analysis fall short even of Damköhler's pioneering work. Hence, determining the turbulent burning velocity remains one of the most important unresolved problems in premixed turbulent combustion. [\[67, 68\]](#)

$$\frac{s_T}{s_L} = 1 + C \left(\frac{v'}{s_L} \right)^n \quad (2.13)$$

2.2.3. FLASHBACK IN PREMIXED COMBUSTION

As mentioned previously under [Subsec. 2.1.2.4](#) flashback risk increases for hydrogen enriched mixtures. Due to the fuel and air being premixed before being injected in the combustor zone the danger of flashback, i.e. that the flame travels upstream into the injector and can cause severe damage, is prevalent. It occurs when the turbulent flame speed exceeds the velocity vector which points in the same direction as the flame speed vector. According to [Lieuwen *et al.* \[69\]](#) and [Reichel \[5\]](#) flashback in swirl stabilized combustors can be subdivided into five categories. These will be briefly covered in the next subsection, together with possible mitigation measurements.

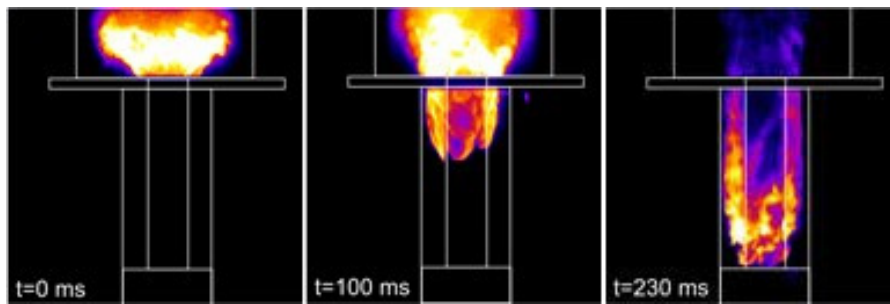


Figure 2.19: LES visualisation flashback phenomena [70]

TYPES OF FLASHBACK

The limits of operational ranges for swirl stabilized combustors were extensively researched by [Huang and Yang \[71\]](#), [Lieuwen *et al.* \[69\]](#) and [Benim and Syed \[56\]](#). [Lieuwen *et al.* \[69\]](#) defined four different types of flashback. A fifth mode of flashback due to autoignition is added to this list by [Reichel \[5\]](#), who experimentally researched the phenomenon, and [Sommerer *et al.* \[72\]](#), who studied the phenomenon using large-eddy simulations (LES). An overview of the modes is provided in [Fig. 2.20](#) below and each mode will be covered in the subsequent subsections.

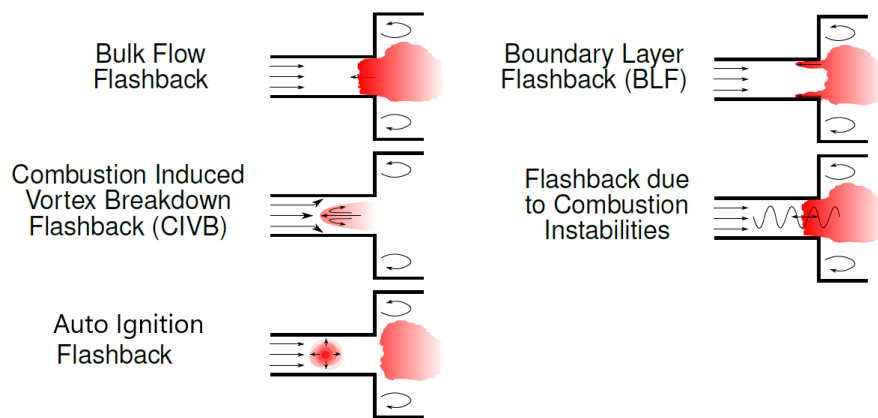


Figure 2.20: Visualisation of flashback modes (Adapted from [73])

Flashback in Core Flow

This first type of flash back is the most straightforward and general as it occurs when the turbulent flame speed exceeds the bulk flow velocity. In this condition the flame is able to move towards the injector given the net velocity difference. This can easily be prevented by ensuring in the combustor design that the bulk flow velocity is sufficiently high as result this mode is often not critical. However, it may still occur along the vortex axis in case of poor flow fields in the combustor, which might stem from vortex breakdown in the premixing tube for example.

Flashback due to Combustion Instabilities

The second type of flashback can be attributed to local flow fluctuations caused by combustion instabilities. Combustion instabilities and their causes are discussed in more detail in [Subsec. 2.5.1](#) but to summarize they are driven by unsteady heat release that causes large pressure oscillations. If these pulsations are large enough they will affect the flow velocity locally causing the flow velocity to periodically drop below the time averaged bulk velocity and generate large scale vortices. Given a low enough frequency of these pulsation the flame will propagate upstream and flashback. Little can be undertaken to combat flashback when these high pulsation instabilities are present in the flow. However, as thermoacoustic instabilities with large magnitudes should be avoided for regular stable combustion this mechanism should not poses any problems during nominal testing.

Wall Boundary Layer Flashback

The next flashback type can be attributed to a fundamental physical aspect of confined flows through pipes. Internal flow physics dictate that a boundary layer is formed alongside the wall due to friction and no slip condition at the wall itself as previously introduced in [Subsec. 2.2.5.3](#). In addition one can also speak of a thermal boundary layer as the flow will be cooled by the walls. This heat loss is also different on the core flow conditions, where the improved mixing properties of the turbulent flow will in fact result in a greater heat loss. This phenomenon will reduce the quenching distance towards the wall and will together with reduced velocity prevent the oxidation reaction to take place close to the wall.

$$g_f \sim \frac{S_L}{\delta_q} \quad (2.14)$$

Wall boundary layer flashback is associated with the a certain metric called the critical velocity gradient (g_f), defined by [Eq. 2.14](#), being exceeded. The flashback occurs when the burning velocity exceeds the flow velocity outside the quenching distance, i.e. the critical velocity gradient is exceeded by the flame velocity. In other words the flashback occurs in the boundary layer due to the reduced local flow velocity yet it only occurs at a reasonable distance from the wall due to the quenching effect. Given the laminar flame speeds and quenching distances provided earlier in [Table 2.1](#) one finds that the velocity gradient of hydrogen is approximately more than 20 times greater than that of methane due to its small quenching distance and high flame speed. This means that hydrogen mixtures are more susceptible to wall boundary layer flashback than methane.

By keeping the boundary layer as thin as possible one reduces the chance of flashback as the BL region outside the quenching distance will be smaller and the flow velocity in it should be higher. Similarly, one should avoid separation as this will move the boundary away from the wall and its quenching effect and the possible induced vortices might even further reduce the flow speeds or cause flow reversal within it. [\[69\]](#) Alternatively, one could add dilution holes in the chamber wall allowing additional air to enter the chamber and effectively making the near-wall region leaner in equivalence ratio. This will cause the flame velocities to decrease and the quenching distance to increase inside this diluted region effectively reducing the velocity gradient. This technique is commonly used in engine design as it also lowers the heat load on the wall. The effect of changing the size and inclination of these holes was experimentally studied by [Baumgartner et al. \[74\]](#).

Flashback due to CIVB

Combustion Induced Vortex Breakdown (CIVB) is a flashback type that is specific to swirl-stabilized flames due to their reliance on Precessing Vortex Core (PVC) and vortex breakdown (VB) for stabilisation which will be introduced later together with swirling flows in [Subsec. 2.2.5](#). The CIVB mechanism finds its origins in a feedback loop associated with VB. Even though the VB takes place downstream of the mixing tube expansion in the main chamber its chemical reaction can lead to a breakdown of the flow further upstream causing the flame to propagate further towards the mixing tube. The loop is stimulated by the interaction between released heat from combustion and the pressure gradient. Eventually, this will result in the VB bubble to be sucked into the premixing, as shown in [Fig. 2.21](#), where the mechanism is reinforced due to the high volume specific heat release.

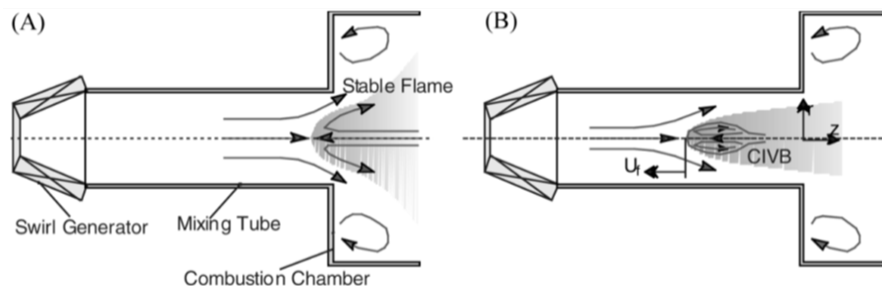


Figure 2.21: Visualisation of CIVB [\[69\]](#)

CVIB Can be combatted by ensuring that the initial VB sits sufficiently aft of the premixing tube exit such that the feedback loop is to weak to propagate the flame upstream. One way to achieve this is through axial

air injection (AAI) which will increase the flow velocity through the center line and push back the central circulation zone. Given its importance with regards to this thesis as the experimental set-up possesses AAI the topic will be further detailed in [Subsec. 2.2.6](#).

Flashback due to Autoignition

The last type of flashback differs from the aforementioned ones in the sense that in these the flashback originates from (elements of) the initial stable flame within the combustion chamber shooting up the flow into the premixer; whereas for this flashback mode the mixture auto-ignites in the pre-mixing tube. autoignition is the spontaneous ignition, i.e. no need for additional igniter like a sparkplug, ect. , of the combustible fuel-oxygen mixture. This will happen when the conditions, being high temperature (600 - 900 K) and high pressure (5 - 35 bar) , are met and the autoignition delay time is shorter than the residence time of the mixture in the premixing tube.

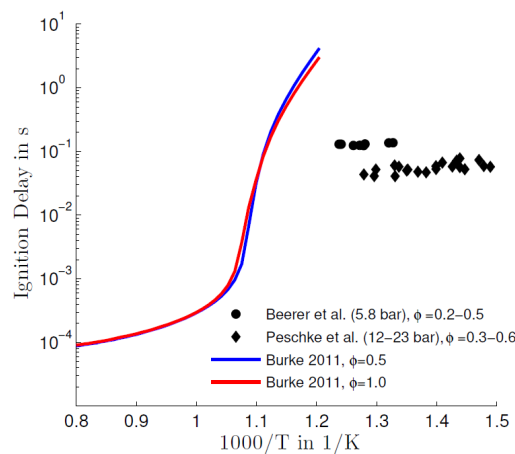


Figure 2.22: autoignition ignition delay times calculated using cantera by [Beer and McDonnell](#)

The autoignition delay times of hydrogen and methane are presented in [Fig. 2.22](#). It is clear that hydrogen has a much shorter delay time, which can be attributed to its high diffusivity, than methane meaning that the risk of autoignition is more elevated. However, a straightforward method to prevent flashback due to autoignition is ensuring that the residence time within the premixing tube is shorter than the autoignition delay time. This can easily be done by shortening the tube or increasing the flow velocity as is the case for the previously mentioned axial air injection.

2.2.4. FLAME BLOWOUT, BLOWOFF AND EXTINCTION

Flame blowoff and blowout are closely related and the later can be seen as a more drastic form of the former. Blowoff simply means that the dynamic imbalance in the flame speed and the flow velocity. When increasing flow velocity for an attached flame the flame will begin to lift-off. However, this will decrease the heat loss rate to the injector which will affect the burning velocity at the base of the flame as shown in [Fig. 2.23](#). At one point the flow velocity will exceed the flame speed across the entire flame and it will blow off. Note that the flame structure remains intact when the flame is blown off. With flame blowout the velocity of the flow plays a greater role. The flow velocity simply trumps the burning velocity of the lifted flame causes it to blow out of the combustor as illustrated in [Fig. 2.24](#).[\[76\]](#)

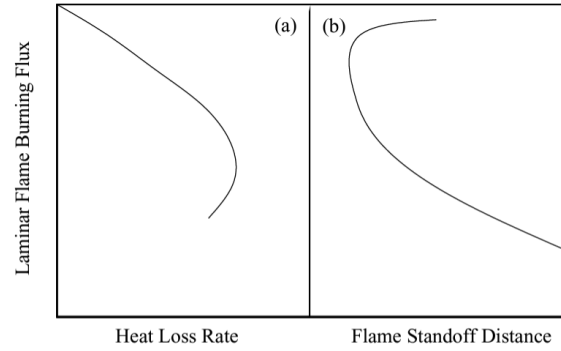


Figure 2.23: flame speed: (a) for a given heat loss rate to the burner surface; (b) for a given flames standoff distance.[76]

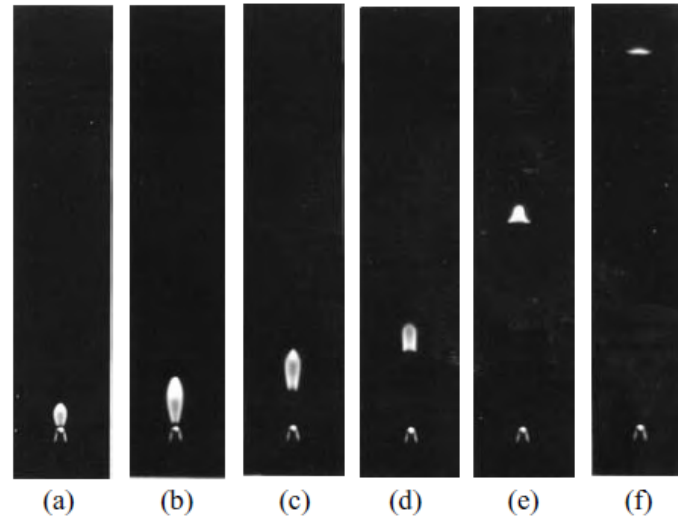


Figure 2.24: Photographic images of attached and lifted flames of propane jets in air showing the blowout mechanism [76]

Flame extinction on the other hand pertains to strong drop in chemical reactivity in the flame which will cause the flame to die out. This can be caused by the mixture being either too lean or too rich meaning that there is either not enough fuel in case of the former or air in case of the latter to sustain the reaction.[76]

As the experiments that will be conducted using equivalence ratios well clear of these ultra-lean/rich conditions this is not a concern for this set-up. Additionally, the addition of hydrogen to methane mixture has shown to extend the lean blowout limits. [77] Therefore the topic of lean blow-out will not be further explored.

2.2.5. SWIRLING FLOWS

Swirl stabilisation, be it through an axial swirler or tangential injection, uses the effects of the induced vorticity on the flow to create a stable flow field wherein the flame can anchor. As swirl increases a low velocity region is created at the center of the swirling flow wherein the flame can stabilise as shown in Fig. 2.25. Above a critical swirl number an Inner or Central Recirculation Zone (CRZ) is formed. The flow reversal that lays at the base of the formation of the CRZ will also supply heat and active species back to the root of the flame. The topic of swirl quantification will be covered in Subsec. 2.2.5.1 together with a brief overview of the different swirl regimes. The highly swirled flow structure that causes the formation of the CRZ will be elucidated in Subsec. 2.2.5.2 together with relevant mechanisms. The role of swirl decay and boundary layer growth will be covered in Subsec. 2.2.5.3 due to their relevance to the formation of the CRZ. Lastly, the Influence of that the swirling flow has on the combustion is discussed in Subsec. 2.2.5.4.

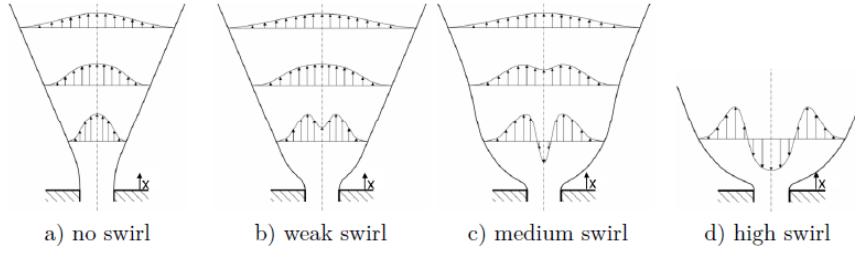


Figure 2.25: Axial velocity profiles for different swirl levels [5]

QUANTIFICATION OF SWIRL

As mentioned previously the amount of swirl plays a critical role in the creation of the vortex breakdown bubble which will generate the CRZ. Therefore, it is important to be able to quantify the swirl such that the injector and chamber can be designed to ensure adequate swirl levels. This can be accomplished by calculating the swirl number of the flow which is defined in the paragraph below. The subsequent paragraph will divulge in the different swirl regimes.

Swirl Number

The swirl in a (pipe) flow can be quantified using the dimensionless swirl number which is the ratio of the axial flux of tangential momentum (G_θ) over the axial flux of axial momentum (G_x). The relation is presented by Eq. 2.15 where r_0 stands for the outer radius. It effectively relates the preference of the flow to rotate radially to the preference to move axially.

$$S_w = \frac{G_\theta}{r_0 G_x} \quad (2.15)$$

The axial flux of the tangential moment and axial thrust are detailed below.

$$G_\theta = \int_0^R (\rho u_\theta r) (u_x 2\pi r) dr \quad (2.16a)$$

$$G_x = \int_0^R (\rho u_x) (u_x 2\pi r) dr + \int_0^R p 2\pi r dr \quad (2.16b)$$

It is important to note that this swirl will be affected by the boundary layer that develops in a confined flow. The topic will be further discussed in Subsec. 2.2.5.3, but in short the swirl will not be able to maintain its radial pressure gradient if the boundary layer grows to envelope the entire width of the chamber and remove the inviscid region. [78]

Swirl Regimes

Swirling flows can be subdivided into two categories, Low and High Swirl, as shown in Fig. 2.26. It is clear that swirl intensity in the low regime is inadequate to trigger the (stable) bubble vortex breakdown which creates the CRZ. In between the upper boundary of 0.5 for the low regime and the lower boundary of 0.6 of the higher regime a transitional or critical regime can be identified. [71] In this regime the vortex breakdown starts to develop and other types vortex breakdown apart from the bubble variant can be perceived. The reason this transitional regime was created is to allow for a clear distinction point between the low and high regimes as the actual transition point differs from design to design as it is dependant on combustor geometry and Reynold's number. The different regimes will be elucidated below.

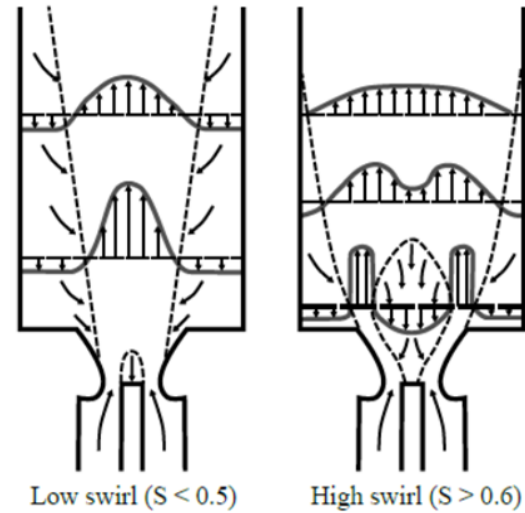


Figure 2.26: Axial velocity profiles for different swirl levels [79]

Low swirl regime: $S_w < 0.5$

For low swirl flows no inner recirculation zone is present as either the swirl and/or turbulence intensity and/or combustor geometry is not sufficient to cause vortex breakdown. Therefore only the corner bubbles that are caused by the sudden expansion of the chamber are present.

Transitional swirl regime: $0.5 < S_w < 0.6$

In this transition phase the swirl intensity alone is not enough to cause stable bubble vortex breakdown but needs to be supported by chamber geometry and turbulence. Therefore several other vortex breakdown modes can be noticed. In pipe flows however the most common breakdowns are the spiral, Fig. 2.27, and the familiar bubble breakdown, Fig. 2.28. The outer recirculation zone still stays present if there is a sudden expansion in the chamber. [71]

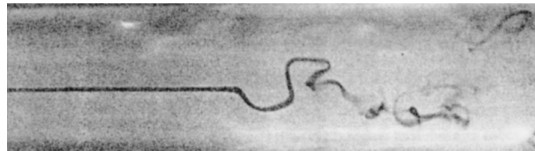


Figure 2.27: Spiral vortex breakdown [80]

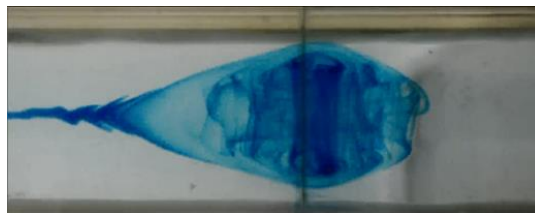


Figure 2.28: Bubble vortex breakdown [81]

High swirl regime: $S_w > 0.6$

The high swirl regime corresponds to the flow regime presented earlier in Subsec. 2.2.5.2. There is a persistent bubble vortex breakdown that generates the CRZ. The flow reversal leads to a significant dip at the core of the flow in the axial velocity profiles and brings back heat and radicals to the root of the flame.

An overview of the different regimes is given in Table 2.2.

Table 2.2: Classification table for swirl regimes

Swirl Regime	Swirl Number	VB type	Pressure Loss
Low	< 0.5	None	Low
Transition	$0.5 < S_w < 0.6$	Spiral or Bubble	Medium
High	> 0.6	Bubble (possibly with PVC)	High

FLOW STRUCTURE IN A HIGH SWIRL FLOW AND MECHANISM

The flow structure of a highly swirled flow is presented in Fig. 2.29, with the CRZ clearly visible. Next to the the CRZ an Outer Recirculation Zone (ORZ) is also indicated together with the shear layers. A shear layer is a thin region separating two different flows with significantly different properties such a speed and possess high velocity gradients and high viscous shearing stresses. The ORZ is formed as a result of the sudden expansion of the flow into the chamber, creating a low-pressure zone in the corners. This low-pressure zone causes the higher-pressure air from the bulk flow to circulate back. Another element indicated in the figure is the Precessing Vortex Core (PVC) within the inner shear layer is not always present. The instability is discussed further in the next subsection but its occurrence has a significant effect on the thermoacoustic instabilities.

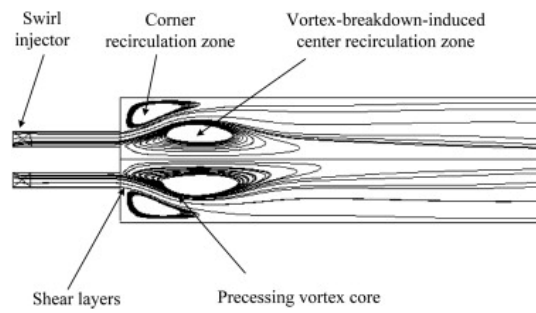


Figure 2.29: Flow structures of a typical gas turbine combustor with a coaxial injector. [71]

The CRZ is formed through a process called vortex breakdown, more precisely it is a bubble vortex breakdown of the core swirling jet located at the centre of the chamber. Other types of vortex breakdown in pipe flow exist at lower swirl intensities and are covered in the next subsection. Due to the high swirl a lower pressure zone is formed at the center of the swirl due to the compressibility of the flow, which will create an adverse axial pressure gradient, forcing the flow to create a negative axial velocity zone. This is enhanced by the sudden expansion of the chamber which will make the CRZ larger. The reason the downstream flow increases in pressure can be attributed to boundary layer growth and swirl decay encountered by the expanding of the core swirling jet which will be further discussed in Subsec. 2.2.5.3. The reversed flow at the centre of the CRZ will bring back radicals and heat to the root of the flame enabling flame stabilisation even in lean conditions. Additionally, the lower velocities found in it will allow for flame anchoring at the locations where the flow speed matches the turbulent burning speed of the flame.

The Precessing Vortex Core

The Precessing Vortex Core(PVC) is single-helical self induced hydrodynamic instability that might occur in the shear layer, which is indicated in Fig. 2.29. In non-reacting conditions the PVC formes above a critical swirl number. In the case of a flame the instability is less predictable and can be suppressed depending on changes of steam content and preheat temperature, swirl number, thermal power, pilot flow rate, or axial air injection.[82] Its occurrence is noteworthy with regards to the flame as it has been strongly linked to transition in flame shapes. As will be discussed in Subsec. 2.2.5.4 any disturbances in the flow have a major impact on flame dynamics, hence so will the PVC.

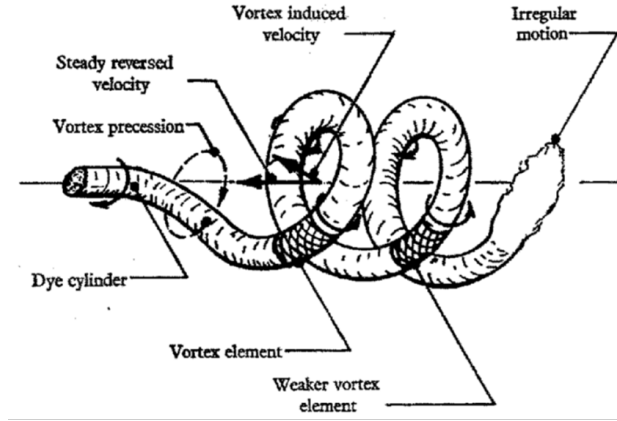


Figure 2.30: The periodic motion of the precessing vortex core of a swirling flame[83]

The PVC, shown in Fig. 2.30, originates in the region of absolute instability in the inner shear layer between the swirling jet and the CRZ. Once established it might grow into the outer shear layer after which it will start imposing its frequency, which can be related to mass flow, on the entire flow field. The relation with acoustic instability is still unclear as experiments have shown to both suppress and enhance them. [82]

SWIRL DECAY AND BOUNDARY LAYER GROWTH

As already alluded to several times in this section, swirl decay and boundary layer growth play an important role in the mechanism of vortex breakdown and are therefore an inherent part of the formation of the CRZ in high swirling flows. The boundary layer is formed when a flow passes over a surface due to the friction and viscous effects that occur at the wall. Due to non-slip condition at the wall there will be a tangential velocity gradient in a small region around the wall as shown in Fig. 2.31. Naturally, the formation of the boundary layer will depend on the bulk flow conditions and whether or not it is laminar or turbulent. The laminar flow sees a more gradual decrease in axial velocity towards the wall compared to the turbulent flow.

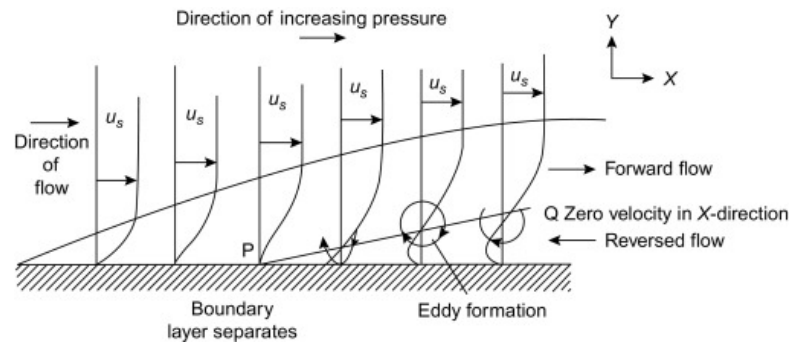


Figure 2.31: Wall boundary layer velocity profile [84]

Not only is there a velocity boundary layer, but there is also due to the presence of the colder wall a thermal boundary layer that characterises the heat loss to the wall in the flow. The boundary layer width of both the thermal boundary layer (δ_T) and the velocity boundary layer (δ) are in fact related through the Prandtl number ($Pr = \nu/\alpha$) which is the ratio of the momentum diffusivity over the thermal diffusivity. In case the $Pr = 1$ both boundary layers will possess the same width in case of a Prandtl number smaller than 1 the velocity BL width will be smaller than that of the thermal ($\delta < \delta_T$).

The formation of these boundary layers will naturally have an effect on the dynamics of the confined swirling flow. Especially, past the entrance length (L_e), which is the length after which the boundary layer traverses the entire width of the pipe. This entrance length is hence an important factor in swirl decay and follows from the boundary layer growth which is characterised by several factors.

Boundary Layer Growth in Pipe Flows

The different velocity boundary layer regimes in pipe flows are presented in Fig. 2.32 and further specified in Table 2.3. The thermal boundary layers develop similarly in pipe flows. Once fully developed these time-averaged structures will always be there regardless of upstream swirl.

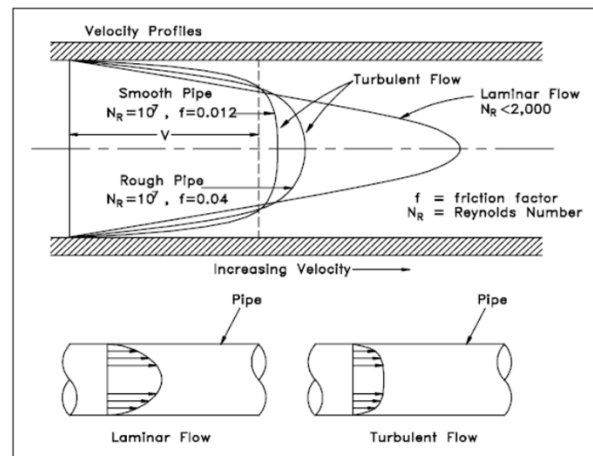


Figure 2.32: Overview of boundary layer development in pipe flows for different flow regimes [85]

Table 2.3: Specifications for boundary layer development in pipe flows at different flow conditions

Flow Regime	Reynolds Number (Re_D)	Entrance Length (L_e/W)
Laminar	$Re_D < 2000$	$L_e/W \approx 0.06 \cdot Re_D$
Transitional	$2000 < Re_D < 4000$	Depends on critical point
Turbulent	$Re_D > 4000$	$L_e/W \approx 4.4 \cdot Re_D^{1/6}$

It is noteworthy that regardless of the flow regime the boundary layer will eventually stretch across the entire width of the pipe. The resulting eddies in this will greatly enhance mixing and make the flow and its velocity more uniform. This is detrimental for the radial pressure distribution that a swirling flow possesses and hence the swirl will decay. As a matter of fact Greitzer *et al.* [78] states that an inviscid core needs to be in place, as shown in Fig. 2.33, in order for the confined swirl to survive. The displacement thickness (δ^*) which grows axially with the boundary layer width will effectively reduce the effective width of the pipe: $W_{eff} = W - 2 \cdot \delta^*$. Given the earlier notion that the boundary layer will eventually encompass the entire width of the pipe it is fair to say that boundary layer growth plays a fundamental role in swirl decay.

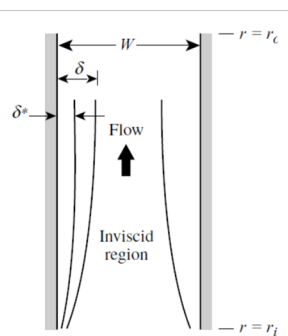


Figure 2.33: Boundary layer effect on effective width in a pipe [78]

Swirl Decay in Pipe Flows

It is both the effect from the thermal and the velocity boundary layer growth that disrupt the swirl in the inviscid core and will eventually grow to the entire width of the pipe. The energy that is taken away by the

wall through the thermal boundary layer and the disruptive eddies originating in the velocity boundary layer will cause the vortex break down as shown for a spiral vortex breakdown in Fig. 2.34.

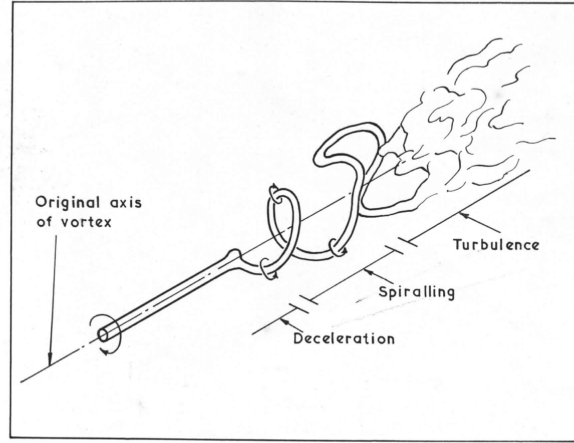


Figure 2.34: Overview of spiral vortex breakdown and its decay [86]

The decay rate has been further researched by Steenbergen and Voskamp [87] who proposed a simple exponential decay rate given by Eq. 2.17.

$$\frac{d(S_w)}{d[L_e/W]} = -\beta \cdot S_w \Rightarrow \frac{S_w}{S_{w0}} = e^{-\beta \cdot L_e/W} \quad (2.17)$$

This decay rate (β) is naturally dependant on the initial swirl number as it is a measure of swirl intensity and the Reynolds number for its influence on boundary layer growth. The values for the decay rate are tabulated in Table 2.4 for the regions of interest. The findings are in line with the earlier made statements: The higher turbulent flow, and thus longer it takes for the boundary layer to encompass the entire flow the slower the swirl decay. Likewise, the the higher initial intensity of the swirling flow the less it is affected by the boundary layer growth.

Table 2.4: Swirl decay rates for different Reynolds numbers and swirl numbers from [87] adapted by [47]

Regime	$0.5 < S_{W0} < 0.8$	$0.8 < S_{W0}$
$2 \times 10^4 < Re_D < 4 \times 10^4$	$\beta \approx 0.037$	$\beta \approx [0.05 - 0.06]$
$1.5 \times 10^4 < Re_D < 2 \times 10^4$	$\beta \approx 0.043$	$\beta \approx [0.05 - 0.07]$

Thus far only non-reacting swirling flows have been discussed. Naturally, the combustion reaction will have a great effect on the behaviour of the flow. This will be discussed in the next subsection.

CONSIDERATIONS FOR TURBULENT PREMIXED COMBUSTION IN SWIRLING FLOW

Apart from the notion that several modes of turbulent premixed combustion can occur at once in a flame due to the variance in dissipation a additional comment should be made with regards to the addition of swirling motion to the flow. Due to the additional dynamics at play that stem from the swirling motion like the vortex breakdown and inner and outer circulation zones. The effect that these disturbances have on the classical premixed turbulent combustion modes presented in previous sections can be quite significant as it will trigger instability effects. One of the common instabilities caused by swirling flames are thermo-acoustic in nature. The sound waves induced by pressure fluctuations which can stem from several sources. For example they can be attributed to periodic motion of the PVC can resonate with combustion chamber structure. These vibration have shown to be detrimental for the structures fatigue and causes an increase in emissions. The topic of combustion instabilities further discussed in Sec. 2.5.

In conclusion, one should sees the earlier presented Borghi diagram, Fig. 2.17 and the included modes as a fundamental tool to help understand and analyse the possible flame regimes in swirling combustion.

2.2.6. AXIAL AIR INJECTION

Axial Air Injection sees the addition of an axial hollow tube on the center line through the swirler as depicted in Fig. 2.35. This will allow air to be directly injected axially in the premixing tube without obtaining any deflection of the swirling vanes. In the premixing tube this non-swirling inner flow will increase the overall bulk axial velocity of the mixture by mixing with the swirling outer layer and in the mean time obtain a swirling motion induced from the same outer layer. This increased axial velocity will increase the flashback resistance of the burner in general and will specifically push the location of the stagnation point of the vortex breakdown further down stream. A downside is that the swirling momentum and thus mixing will be reduced and the lean flame blowout limit is increased. A comparing normal and AAI flows is presented in Fig. 2.36.

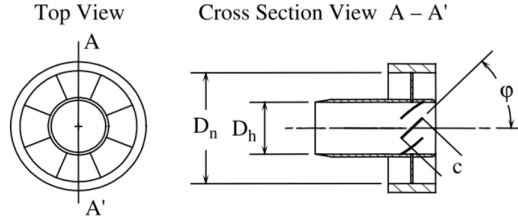


Figure 2.35: Sideview of swirler with hollow AAI tube at the centerline [71]

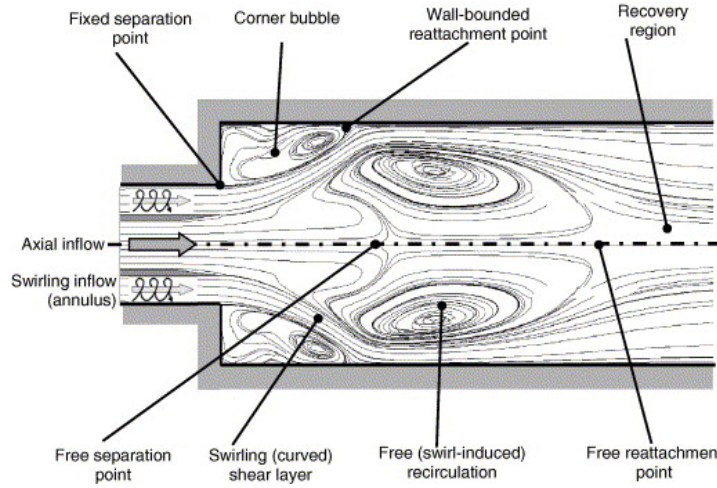


Figure 2.36: Effect of AAI on flow structure [88]

The swirl number can now be quantified using Eq. 2.18 which takes into account the added effect of the central axial air stream. Radius of the inner axial air stream is represented by R_h , while the outer radius of the swirl is given by R_n . The tip vane angle ϕ_0 is also of importance. [89]

$$S_w = 0.5 \frac{1}{1 - \psi} \frac{1 - (R_h/R_n)^4}{1 - (R_h/R_n)^2} \tan(\phi_0) \quad (2.18)$$

The blockage factor, ψ , indicate how much the swirler vanes impeach the axial flow. It can be calculated using Eq. 2.19, for helical swirler this results in Eq. 2.20 where the projected area of the vane is defined integral of the locally projected thickness times the radius step which is dependant on the local vane angle.

$$\psi = \frac{A_{\text{vane}} \cdot n_{\text{vane}}}{(R_t^2 - R_h^2) \pi} \quad (2.19)$$

$$\psi_{\text{hel}} = \frac{n_{\text{vane}} \int_{R_h}^{R_t} \frac{t}{\cos(\phi)} dr}{(R_t^2 - R_h^2) \pi} \quad (2.20)$$

It is also important to denote how the AAI is quantified and more importantly varied as this may differ between sources. For the purpose of this thesis the AAI will be quantified by its mass flow relative to the total mass flow as shown in [Eq. 2.21](#).

$$\chi = \frac{\dot{m}_{AAI}}{\dot{m}_{total}} \quad (2.21)$$

2.3. RELEVANT DIAGNOSTIC TECHNIQUES

The field of experimental combustion research is heavily reliant on a diverse array of diagnostic techniques to gain essential scientific insights. These diagnostics serve as essential tools for quantifying parameters such as temperature, velocity, species concentrations, flame structure, and emissions. There exists only a few optical and non-optical techniques that measure these various quantities, each with their own specifications, measurements and drawbacks.

Optical techniques represent a fundamental component of experimental combustion research. They provide researchers with valuable spatial and temporal information about critical aspects of combustion, including chemical species distribution, temperature profiles, and flame dynamics. The most common optical techniques are laser-induced fluorescence (LIF), absorption spectroscopy, laser Doppler anemometry (LDA), and tracing particle as used in Particle Induced Velocimetry (PIV).

On the other hand, non-optical measurements are equally essential in gaining a comprehensive understanding of reacting flows. These measurements include parameters such as pressure, temperature, and gas composition. Pressure transducers, temperature probes and gas analyzers are commonly used non-optical diagnostic tools in combustion research. Pressure measurements help researchers study the thermoacoustics of combustion, while temperature measurements are crucial for monitoring the system and establishing the heat transfers. Gas composition data are essential for characterizing the combustion products and assessing emissions, which is vital for environmental considerations.

In practice, a combination of optical and non-optical measurements is often used to gain a comprehensive understanding of reacting flows. Optical techniques provide valuable spatial and temporal information about chemical species, flame location, and flame dynamics, while non-optical measurements offer critical data on pressure, temperature, and gas composition. In this chapter all types of measurements taken in the experiments will be covered and the exact techniques that will be used will be highlighted. First concentration measurements are discussed in [Subsec. 2.3.1](#), followed by velocity measurements in [Subsec. 2.3.2](#). Then Acoustic measurements are briefly covered in [Subsec. 2.3.3](#) due to their importance for thermo-acoustic instability research. Lastly, pressure and temperature measurement techniques are elucidated in [Subsec. 2.3.4](#).

2.3.1. CONCENTRATION MEASUREMENTS

Species concentration measurements are not only used by researchers to analyse the species apparent in the flame and emissions. By cleverly relating the concentration of certain species to the fundamental knowledge of their occurrence one can also gain insights regarding temperature and flame location.

There exist multiple methods to study these aspects, ranging from the non optical probe sampling, covered in [Subsec. 2.3.1.3](#), to Laser-Induced Fluorescence (LIF) where certain particles are excited using a laser as detailed in [Subsec. 2.3.1.2](#). The set-up used in the thesis also includes chemiluminescence which is discussed in [Subsec. 2.3.1.1](#). Other common techniques not employed in the setup include infrared and Raman spectroscopy, which utilizes laser light scattering to provide information about molecular vibrational energy levels, enabling the identification and quantification of species concentrations within the flame; and Coherent anti-Stokes Raman (CARS) spectroscopy, an advanced laser-based technique that generates a coherent anti-Stokes signal to measure molecular vibrations, offering precise measurements of species concentrations and temperature in combustion processes and other applications.

Naturally, the characteristics of each technique will determine what type of species it is able to detect. In this light an important distinction can be made between minor and major species. This distinction relates to the relative concentration of the species, i.e. major species, like CO_2 and H_2O , occur in high concentration in the flow and make out the bulk of the emissions. Minor species, like NO_x , CH and CH_2OH , occur in far smaller quantities and can only be captured by techniques that are refined enough. For example, gas analysers and infrared spectroscopy are used for major species detection whereas more sensitive techniques like Raman and CARS spectroscopy, PLIF and chemiluminescence are used towards the measurement of minor species.

CHEMILUMINESCENCE

Chemiluminescence is a straightforward yet powerful non-intrusive measurement technique that revolves around capturing and characterizing the light emitted by a flame. At its core, chemiluminescence relies on the connection between light emissions across certain wavelengths and the occurrence of certain (intermediate) species. Within the combustion reaction certain chemical species undergo transformations, resulting in the formation of excited-state intermediates or products. These excited species release energy in the form of photons, giving rise to the emission of visible or ultraviolet light. Some emissions can be coupled to the occurrence of certain species due to the narrow bandwidth over which they occur as shown in Fig. 2.37. This property enables researchers to gain insights into the concentration of specific species that can be used to fundamental aspects of flame behavior like flame front location and (qualitative) temperature indications.

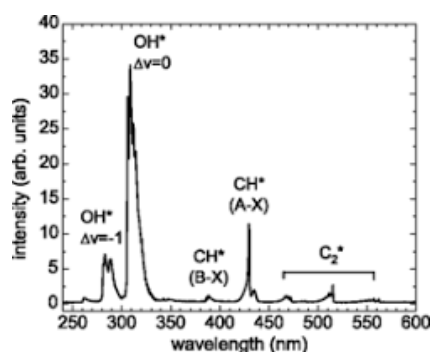


Figure 2.37: Chemiluminescence spectrum of a CH₄/O₂ flame with 30 % H₂ addition [90]

This principle gives way to two different philosophies and applications, as indicated in Fig. 2.38. Either one can opt to use a bandpass filter and intensifier to focus on a narrow bandwidth and single out a specific species to study it closer. More specifically, bandpass filtered chemiluminescence is a spectroscopic technique that isolates a narrow range of wavelengths from the emitted light by the combustion. It uses a bandpass filter to selectively transmit light of interest while blocking out other wavelengths. This method is valuable when one knows the chemiluminescence specific wavelengths of a species of interest. By applying the filter one can visualise the concentration of said species in the reaction zone through the intensity of the emitted light. It is particularly useful for targeted analytical assays and studies focused on specific chemical species emitting at distinct wavelengths. For example, given the spectrum shown in Fig. 2.37 one can easily visualise the OH^{*} ($\lambda = 310\text{nm}$) given its high intensity and narrow wavelength band. [91] In fact OH^{*} is typically used to identify the flame front and temperature within the flame given that it is a shortly naturally excited in the flamefront. On the other hand, for higher carbon containing fuels CH^{*} ($\lambda = 388, 431\text{nm}$) is also often investigated, given that it is a short lived radical that is only formed at final stages of fuel decomposition. This allows one to identify the flame front or stoichiometric surface in premixed or non-premixed flames respectively. One should note that it is possible to combine filters with an image doubler to study multiple bandwidths at once for the purpose of analysing more complex species. [92, 93]

Spectrally Resolved Chemiluminescence, on the other hand, is a comprehensive approach that captures the entire emission spectrum produced during a chemiluminescent reaction. Instead of isolating a specific wavelength range, it provides a detailed spectral profile of the emitted light by the combustion reaction. For this it uses a spectrograph or monochromator, in case one wants to limit the range of the studied spectrum, in combination with an Intensified Charge-Coupled Device (ICCD) to capture the image over the desired spectrum. This technique is used to visualise the entire emission spectrum, as shown in Fig. 2.38, of the flame allowing one to identify key species and characterise complex reaction pathways. [92]

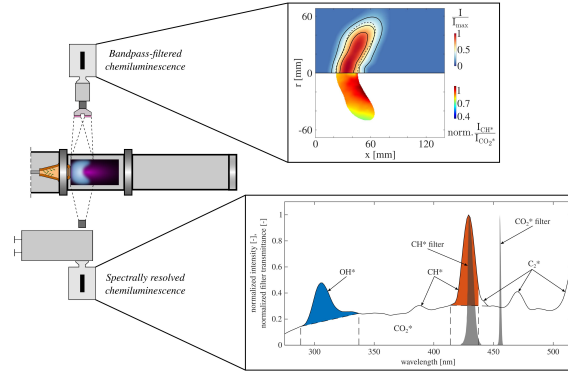


Figure 2.38: Both spectral and bandpass-filtered chemiluminescence application visualised for kerosene burner. [92]

Given that chemiluminescence captures a line-of-sight that is generated from spatial fields. Abel deconvolution plays a crucial role in mitigating the effects of optical blurring and obtain the original symmetric function from it's projection i.e. a scan or a photograph. Abel deconvolution effectively deblurs these images, providing a clearer representation of the flame's spatial characteristics.[91]

LASER-INDUCED FLUORESCENCE (LIF)

Laser-Induced Fluorescence (LIF) is a complexer technique than chemiluminescence as it employs a laser to excite the target molecules or species within a bandwidth and then capturing the fluorescent "mie" scattered light they emit when they return to their ground state. The laser, which can be in the UV or visible spectra, is chosen such that it only targets a certain narrow range of frequencies. For example for OH lasers of $282nm$ or $306nm$ are typically used which is inline with the peaks associated with OH presented in the emission spectrum shown in Fig. 2.37. Example measurements are shown below in Fig. 2.39.

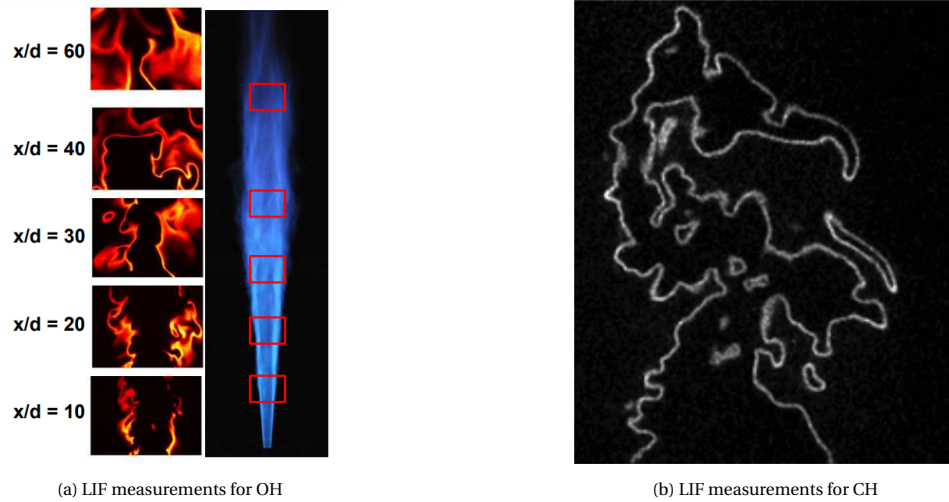


Figure 2.39: Example visualisations of different LIF measurements [93]

Figure 2.39a also shows an important nuance between the OH-LIF images and the images produced by chemiluminescence with naturally excited OH^* emissions. The OH molecules that are excited by the laser lose their energy more slowly as its higher energy is dissipated when the radical is broken down by slow three-body diagram. This makes OH LIF images more adequate to visualise the combustion products, turbulence and temperature⁴ within the flame. [93]

⁴As OH is still only formed in high temperature regions

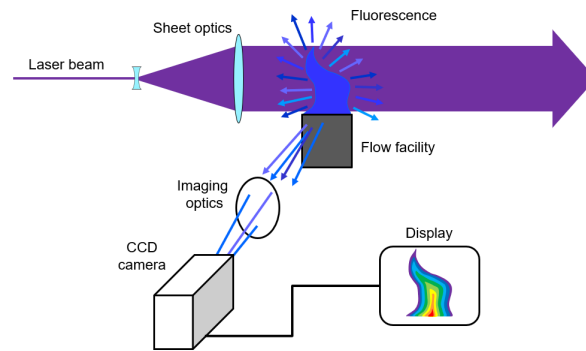


Figure 2.40: Schematic PLIF set-up. [94]

A typical set-up for Planar Laser-Induced Fluorescence is presented in Fig. 2.40. A (pulsed) laser is transformed into a sheet that illuminates the chosen cross-section, this is then captured using an UV or ICCD camera. The pulsed lasers allows for time-resolved measurements to take place⁵ that then can be averaged together. While it is mainly used as a qualitative visualization technique it is possible to quantify the concentrations measurements but it is very hard to achieve in practice. Additionally, research is being conducted to obtain temperature measurements from the OH LIF images. [93]

EXHAUST GAS EMISSION ANALYSER

The exhaust gases also contain a lot of information regarding the flame efficiency and behaviour. Furthermore, the occurrence of certain harmful and toxic gasses is of special interest given the detrimental effects they have on the climate and humans as explained in Sec. 2.2. Therefore, it is wise to capture the exhaust gases and analyse a small sample in the gas analyser.

Typical measurements focus on the quantization of particles like CO, CH₄, CO₂, O₂, NO, NO₂ and soot in the sample. This done via a broad variety of techniques: Gas Chromatography, Mass Spectrometry, Fourier Transform Infrared Spectroscopy and Flame Ionization Detector to name a few. The set-up used in the thesis uses the *Limas 21*, *Uras 26* for CO, CH₄, CO₂ and *Magnos 26* developed and manufactured by ABB. [95]

Uras (NDIR):

- Utilizes Non-Dispersive Infrared (NDIR) spectroscopy.
- Specialized for NO_x and SO₂ measurement in exhaust gases.

Limas (NDUV):

- Relies on Non-Dispersive Ultraviolet (NDUV) spectroscopy.
- Used for detecting gases like CO, CH₄, CO₂.

Magnos (Paramagnetic):

- Based on paramagnetic analysis.
- Primarily measures percentage-level oxygen.

2.3.2. VELOCITY MEASUREMENTS

Velocity measurement techniques are paramount in experimental combustion research, offering precise quantification of fluid dynamics within combustion systems. Various velocity measurement techniques have been developed in experimental combustion. Particle Image Velocimetry (PIV) employs laser-induced light scattering to capture particle motion and determine flow velocities. As this is the most common technique and it will be used in the experiments used in the thesis it will be further discussed in the next subsection. Other common techniques not discussed in detail include Laser Doppler Velocimetry (LDV), which measures velocity by analyzing the Doppler shift in laser light scattered by particles or molecules. Hot-wire anemometry relies on a heated wire to gauge airspeed based on changes in wire resistance.

⁵In the case of high-frequency measurements

PARTICLE IMAGE VELOCIMETRY (PIV)

Particle Image Velocimetry is the most common method to visualise and analyse the flow field as it allows one to capture the instantaneous velocity of the flow. It is a non-intrusive diagnostic technique that tracks certain seeder or tracker particles that are added to the flow. More specifically non-reacting seeding particles, typically titanium dioxide (TiO₂) for high temperature applications, are added to the flow. The light of the laser is mie scattered light and can be captured with a CCD camera. Taking two pictures in quick succession while simultaneously sending out the laser pulses allows one to track the displacement of said particles across the (thin) planar cross-section the laser illuminated. The most basic variant of PIV is the 2D2C set-up presented below in Fig. 2.41. It provides a two dimensional measurement giving one the two in-plane velocity components respectively. [96]

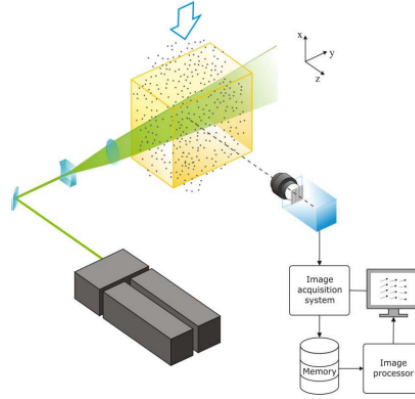


Figure 2.41: Most basic 2D2C PIV set-up [97]

From the displacement one can derive the instantaneous velocity given the short time interval in which the pictures are taken. For this the pair can be divided in small interrogation windows where the particles in these windows can be compared using cross-correlation of their patterns. This cross-correlation tells one what the average displacement of the particles is in the two dimensional plane through the correlation peak as shown in Fig. 2.42. Given the average displacement one can derive the average velocity of the particles in the window knowing the laser pulse time interval. The velocity data is hence indirectly inferred from the measurements. [47, 96, 97]

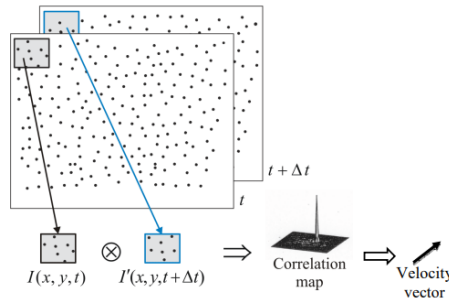


Figure 2.42: Obtaining average displacement in a interrogation window through cross-correlation [97]

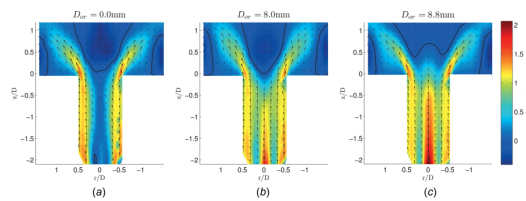


Figure 2.43: PIV results for swirl stabilized burner [5]

By taking several PIV measurements one obtains a large set of instantaneous velocities of the flow. These can be compounded all together to visualise the main flow behaviour and governing dynamics as done in Fig. 2.43. On the other hand a spectral Proper Orthogonal Decomposition (SPOD) can also be used on this time-series data set to visualise and study oscillations and instabilities like the PVC. [96, 97]

Stereoscopic PIV

Stereoscopic PIV or stereo PIV is a more complex PIV set-up that utilises two cameras that are aimed at different angles to the flow. This allows one to create a better view of the displacement in the third plane. A stereo PIV set-up is shown in Fig. 2.44. It is important to note that stereo PIV also just uses one laser sheet meaning that the complication regarding out of plane movement still hold and that the laser pulses and frames need to follow in quick succession.

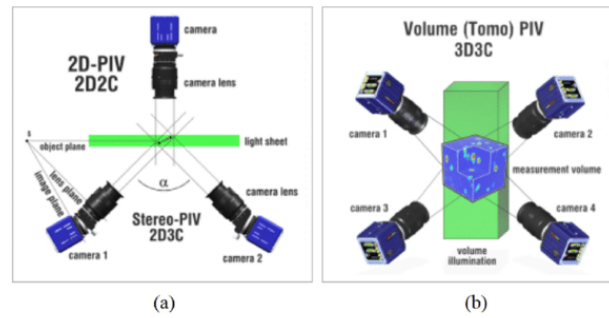


Figure 2.44: Different PIV set-ups [98]

Spectral Proper Orthogonal Decomposition (SPOD)

In the context of PIV measurements Spectral Proper Orthogonal Decomposition [99] is typically used to extract and identify the most significant patterns and structures inherent in complex flow datasets. The technique stems from the classical Proper Orthogonal Decomposition (POD), yet offers a decomposition in the frequency domain allowing one to select more precisely the relevant modes translating to turbulent flow structures. This is achieved by applying a direct Fourier transformation to the data before performing the eigenvalue calculation. By decomposing the velocity field into a set of orthogonal modes as shown in Fig. 2.45, SPOD reveals the dominant spatial and temporal features of the flow. The modes are ordered based on their energy content. The analysis can be used to qualitatively show the occurrence of flow structures like a PVC. [100] An example of SPOD analysis are provided in Fig. 2.55.

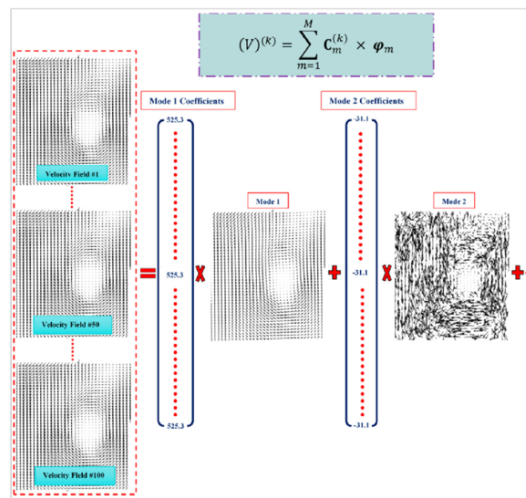


Figure 2.45: Visualisation of spectral POD [98]

2.3.3. ACOUSTIC MEASUREMENTS

Due the detrimental effects that pressure fluctuations, like thermoacoustic instabilities, can have on combustion systems and their link to an increase in emissions it is important to make acoustic measurements during testing to further understand the underlying mechanisms. This can be done by placing a high frequency microphone near or inside⁶ the combustion chamber. Where the former is quite easy to set-up the later is quite challenging due to the required cooling however, the latter does allow for proper thermoacoustic measurements and flame transfer function predictions. The recorded audio then can be processed with an Fast Fourier Transform (FFT) to study the frequency domain of the signal for which an example is given in Fig. 2.46. It is also common to use the Welch method to reduce the variance of the measurement.

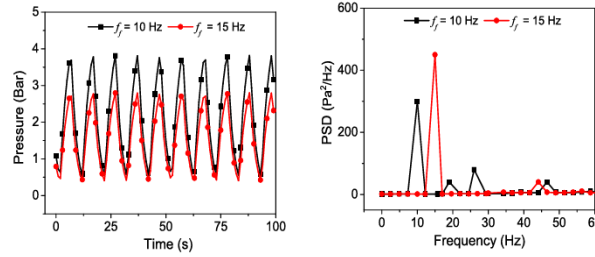


Figure 2.46: (a) Pressure fluctuation and (b) Corresponding FFT analysis. [101]

WELCH METHOD

Welch's method [102] is a widely used technique for estimating the power spectral density (PSD) of a signal. It operates by dividing the data into overlapping segments, applying a window function to each segment to reduce spectral leakage, computing a modified periodogram for each segment using the Fast Fourier Transform (FFT), and finally averaging these periodograms to produce a smoother estimate of the PSD. The method was implemented using the "scipy.signal.welch" function with an overlap of 50%.

2.3.4. PRESSURE AND TEMPERATURE MEASUREMENTS

Set-ups usually also have several pressure and temperature sensors that measure the conditions at several positions in the setup, some near or in the chamber, to ensure safety and proper operations. Typically, several thermocouples and pressure probes are installed for this purpose. These measurements do also pose useful information towards the flame behaviour and dynamics as it can aid to explain certain events such as combustion instabilities.

Thermocouples are temperature sensors that are made up of two different wires. The resistance of the wires changes with temperature and therefore a signal through them can be directly related to the temperature (Seebeck effect). By attaching these in several places the temperature of the set-up can be monitored throughout the experiments.

The pressure probes work by utilizing the piezoelectric effect or the piezoresistive effect, both of which involve changes in electrical properties in response to mechanical pressure. This either a crystal that generates an electrical charge when subjected to pressure changes or resistors whose electrical resistance varies with pressure. These sensors convert the pressure fluctuations within the combustion chamber or fuel/air lines into electrical signals, which are then measured and recorded. The pressure oscillations occurring either the fuel line or the chamber itself can be used to better understand the mechanisms responsible for combustion instabilities. The difference with respect to acoustic probes stems from the range and scales at which both operate. The acoustic sensor can capture frequencies around over broad range whereas the pressure probe is quite limited and used to capture the pressure waves from inside the chamber itself.

However, the practical arrangements for the addition of pressure sensors and thermocouples in the chamber for the set-up that is used for the thesis is still being further researched.

⁶Due to the high temperatures this would need a cooled microphone

2.4. HYDROGEN ENRICHMENT RESEARCH

In this chapter relevant experimental work regarding hydrogen enrichment of swirl stabilised systems by other institutes and predecessors will be covered. Given that the focus of the thesis lies on the research of effect of equivalence ratio oscillations stemming from the fuel system on hydrogen enriched methane flames it is decided to only provide a overview of the most recent work in the field of hydrogen enrichment in this literature study and point the reader to the most recent sources in [Subsec. 2.4.1](#). This section will cover the main takeaways and specifications of the different research that has taken place that is relevant for this thesis and the overview is culminated in [Table 2.5](#).

However, the work of a select group of experiments is highlighted in the subsequent sections due to its close ties to the thesis project. The work by [Reichel](#) is covered in [Subsec. 2.4.2](#) as the used set-up and the concept of AAI employed in it were a big inspiration for the set-up used for the thesis. The work by the IMFT can be regarded as the current state of the art in hydrogen swirl combustion research and will be discussed in the next section, [Subsec. 2.4.3](#). Lastly, the previous work done with regards to the to-be-used set-up [\[47\]](#) will be covered in [Subsec. 2.4.4](#).

2.4.1. OVERVIEW OF STATE OF THE ART EXPERIMENTS

There exists already a lot of literature with regards to methane flames and the stabilisation of them. The stability of and the mechanisms behind have been researched extensively. The dynamics and combustion behaviour of methane flames are therefore quite well understood. For example it has been observed by [Terhaar *et al.* \[103\]](#) that flame shape for swirling methane flames is strongly related to complex flow structures, with the flame vortex interactions becoming more strongly pronounced at leaner conditions. Furthermore, the flame root of lean methane flames is weakened due to the strong flow strain that persists in swirling flows [\[7\]](#). The relative small size of methane, which is in fact smallest hydrocarbon, also limits and simplifies the various reaction pathways that can take place in the combustion reaction. Hence, these flames have also been studied extensively with numerical solvers that include detailed chemical kinetic mechanisms and CFD simulations. In the paper on the LES simulations of hydrogen enriched methane flames by [Ferrante *et al.* \[104\]](#), that are discussed in more detail in [Subsec. 2.4.4.2](#), the methane case is used and compared to experimental results to validate the proposed model and approach.

Compared to methane air flames the flame dynamics and characteristics of hydrogen enriched flames are still not well understood due to the different behaviour the flame possess as a result of the properties attributed to hydrogen as detailed in [Subsec. 2.1.2.2](#). An overview of the various institutes studying hydrogen flames in swirl combustors is provided in [Table 2.5](#) together with some key data on their system as well as the references to some of their latest published work. As is clearly shown in the table, the range of topics being researched is broad and most set-ups differ on several factors. However, the measurement techniques that are used to study the combustion flows are all centered around the core ones covered in the previous chapter, [Sec. 2.3](#), namely acoustics, chemiluminescences, PIV and PLIF.

Most of the set-ups presented in [Table 2.5](#) are equipped with an axial swirler [\[5, 105–113\]](#) like the set-up that will be the subject of this research [\[47\]](#). However, the system utilised by [Guiberti *et al.* \[114\]](#) uses a radial swirler, the researchers at NTNU tested an array of swirlers in an annular combustor [\[115\]](#) and the tests at MIT featured a Silo combustor [\[116\]](#). The concept of Axial Air Injection, which is a key aspect of the work at the TU Delft, is a quite novel concept and can only be found back in the work of [Terhaar *et al.* \[103\]](#) and [Reichel \[5\]](#). Only the test rig used at the IMFT [\[107, 117–119\]](#) utilizes a similar concept where an axial tube allows for axial injection along the center line. However, in their concept hydrogen is injected along the center line. Furthermore, the HY-LON (HYdrogen LOW Nox) combustor at IMFT allows for one to swirl this centrally injected hydrogen creating this so called dual swirl concept. Other concepts [\[7, 109, 120–122\]](#) use a bluff body at the end of the premixing tube to enhance the formation of a central recirculation zone.

Looking at the different equivalence ratios in the chart it is clear that some concepts [\[107, 111, 116, 122\]](#) use the broader flammability limits, that are enabled by the hydrogen, to create ultra lean combustors. These conditions are not tested in the TU Delft combustor due to limitations in the mass flow controllers of the set-up and the relative low pay-off as in the end the combustor needs to deliver the power required. Even fewer of test rigs operate above atmospheric conditions. [\[105, 112, 115\]](#)

Several of the papers [\[105, 122–124\]](#) study the dynamics and flame shapes [\[109, 114, 118, 125\]](#) of hydrogen flames. It is noted that when the hydrogen content or pressure are increased the flame length decreases

and the flame shifts closer towards the injector. More specifically, with the increasing hydrogen content the flame is said to transition from a typical M- or W-shaped flame to transition into a bi-stable flame to end up being a shear layer stabilised V- or bowl-flame. The M-shaped flame relies on stabilisation mechanisms in the central flow with its outer edges being stabilised by the outer recirculations zones. The V-shaped flame is stabilised in the inner shear layer through flame-flow velocity balances [105, 110, 122]. [Chtereov and Boxx](#) [105] noted that a precessing vortex core existed across all the conditions ⁷, however the frequency of it reduces to different Strouhal numbers depending on the flame shape. The PVC has also been studied in greater detail by [Datta et al.](#) [126] for methane mixture with up to 50% hydrogen fuel fraction. In his research, [Datta et al.](#) [126] finds that the higher hydrogen content suppresses the PVC. Furthermore, it is noted in the research that the hydrogen increase also suppresses flame lift-off since the PVC causes local flame extinction near the nozzle center body. In cases where they did notice a PVC it was noted that the PVC develops into the flow allowing intermittent flame reattachment. [Mohammadnejad et al.](#) [127, 128] looks closer into the macro structures within the flame and links the disparity in heat release and local consumption velocities to flame thickening a phenomenon that is said to decrease with increasing equivalence ratio towards stoichiometry and/or hydrogen content. The effect of wall temperature plays also an important role on the flame shape, where the exact location of the flame is then influenced by hydrogen content as shown by the work done by the EM2C lab [129–131]. In the paper by [Guiberti et al.](#) [129] it is stated that the transition between flame shapes is triggered when the tip of the swirl is quenched at the combustor wall side this has been coupled to a critical Karlovitz number.

Flame stability has been studied using a Karlovitz based approach by [Taamallah et al.](#) [132]. In fact the topic of stability plays a major role when hydrogen is added to the mixture and therefore it has been the main topic of several papers [107, 110, 117, 119, 120, 126]. A difference can even be made with regards to studies focused on flashback [5, 112, 113, 133–135] and those focused around thermo-acoustic instabilities [106, 108, 111, 115, 119, 136, 137].

Flashback, which was already identified to play a major role in hydrogen combustion due to the higher burning velocity, is discussed in [Sec. 2.2](#). To combat flashback, the TU Delft set-up includes the installation of an AAI, similar to the approach taken by TU Berlin, as mentioned in [5, 103]. [Reichel](#) [5], in his dissertation [5], dedicated his work to the topic of increasing flashback resistance in premixed swirl stabilised hydrogen-methane combustors. His research is further explored in [Subsec. 2.4.2](#).

The various mechanisms behind thermo-acoustics in the different set-ups is studied in a variety of papers and is subject to the combustor geometry and operating conditions. Therefore a detailed analysis of the set-up of the TU Delft combustor, which operates at atmospheric conditions, needs to be conducted. Therefore it is important to keep in mind that, given that most jet engine combustor operate at increased pressure and have a closed downstream boundary, the acoustics of the engine will change [133]. Interestingly, [Strollo et al.](#) looks into the onset and decay of these instabilities by inducing them into the flow under controlled conditions. The link between thermo-acoustics back to the macro structures within the flame has been studied by [Taamallah et al.](#) [138].

⁷The condition that were researched were hydrogen methane fuel blends with up to 50% hydrogen fraction

Table 2.5: Overview of various institutions researching hydrogen and swirled combustion, originally obtained from Mao *et al.* [7] but extended upon

Institution	ϕ	α_{H_2}	Reynolds Number (Re)	Pressure (atm)	Combustor Info	Sources
DLR	0.69 – 0.71	0% – 50%	30,000 – 100,000	1 – 5	Premixed, Swirl	[105, 123, 126, 136]
EM2C	0.5 – 0.9	0% – 78%	–	1	Premixed, Radial Swirler	[114, 129–131]
ISU	0.7	0% – 55%	8,796 – 9,143	1	Premixed, swirl	[106, 125, 137]
					Tech. premixed	
IMFT	0.4 – 0.9	0% – 100%	5,540 – 15,380	1	Dual Swirler	[107, 117–119]
					Axial H2 Injection	
Imperial	0.55 – 0.7	0% – 40%	19,000	1	Premixed, Swirl	[108]
					Diffuser	
MIT	0.4 – 0.66	0% – 50%	20,000	1	Silo combustor	[116, 132, 138, 139]
NTNU	0.65 – 1	0% – 20%	–	1.5 – 3.3	Annular combustor	[115]
PSU	0.65 – 0.7	0% – 40%	17,000	1	Premixed, Swirl	[109]
					Center body	
UBC/NRC	0.6 – 0.9	0% – 80%	–	1	Premixed Swirl	[110, 127, 128]
UTRC/LBNL	0.3 – 0.4	0% – 90%	–	1	Premixed Swirl	[61, 111, 124]
UT/PSI	0.4 – 0.9	0% – 95%	3,000–40,000	1 – 2.5	Premixed Swirl	[112, 133–135]
Toronto	0.7 – 0.75	0% – 70%	–	1	Pemixed, Swirl	[113]
					Bluff body	
TU Delft	0.55 – 0.9	0% – 100%	10,000–17,000	1	Tech. premixed	[47, 104, 140]
					Axial Air Injection	This work
TU Berlin	0.7 – 0.9	0% – 70%	40,000 – 60,000	1	Tech. premixed	[5, 103]
					Axial Air Injection	
XJTU	0.26 – 0.7	0% – 80%	19,800 – 30,000	1	Pemixed, Swirl	[7, 120–122]
					Bluff body	

Abbreviation	Institution	Abbreviation	Institution
DLR	Deutsches Zentrum für Luft- und Raumfahrt	NRC	National Research Council Canada
EM2C	Laboratoire EM2C	UTRC	United Technologies Research Center
ISU	Iowa State University	LBNL	Lawrence Berkeley National Laboratory
IMFT	Institut de Mécanique des Fluides de Toulouse	UT	University of Texas
Imperial	Imperial College London	PSI	Paul Scherrer Institut
MIT	Massachusetts Institute of Technology	Toronto	University of Toronto
NTNU	Norwegian University of Science and Technology	TU Delft	Technical University Delft
PSU	Pennsylvania State University	TU Berlin	Technical University Berlin
UBC	University of British Columbia	XJTU	Xi'an Jiaotong University

2.4.2. FLASHBACK PREVENTION IN LEAN PRE-MIXED HYDROGEN COMBUSTION (TU BERLIN)

Within the scope of the AHEAD project dr. ir. T. Reichel has published several papers, bundled together in his thesis [5] with respect to his PhD research on Flashback Prevention in Lean Pre-mixed Hydrogen Combustion. Through his publications, he researched on how to increase the flashback resistance of of a LPM combustor with swirl stabilisation by adding axial air injection and the subsequent effects of it. This research includes flow analysis through PIV and flame location analysis with OH^* -chemiluminescence and LIF. The set-up is depicted below in Fig. 2.47. The air and fuel are axially injected into a swirl generator ($S_w = 0.9$) next to the additional axial air injection via a dedicated tube on the center line. These streams meet in a small premixing tube before being dumped and combusted downstream in the cylindrical combustion chamber. The chamber wall was made of quartz glass to allow for laser measurements. The applied techniques where: OH-PLIF, OH^* -chemiluminescence, PIV and NOx detection.

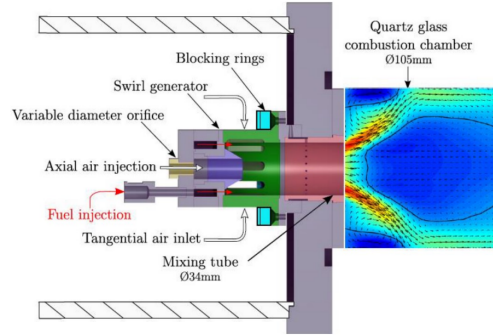


Figure 2.47: Cross-sectional schematic of injector employed for AAI experiments at TU Berlin [5]

The effect on the flow field for two different swirl numbers and AAI was studied using 2D2C PIV for visualisation. Additionally, the flow field for non reacting flows was also studied using water. For the reacting flow field the AAI was varied as a percentage of total airflow. An example of the results is provided below in Fig. 2.48.

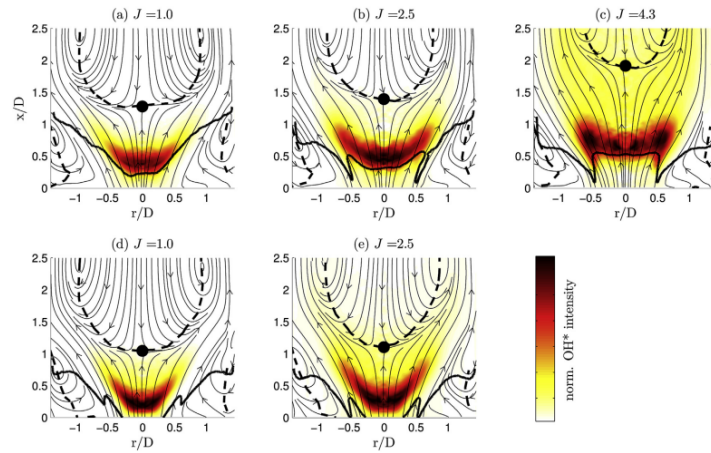


Figure 2.48: OH^* density overlapped with axial velocity data for different momentum ratios in order to visualise flame front and CRZ/ORZ [5]

The momentum ratio referred to in the caption of Fig. 2.48 stands for the fuel-air momentum ratio, Eq. 2.22. It is an important metric as changes in it due to mass flow increases or fuel types can alter the penetration depth of the fuel jets into the air flow and, thus, the fuel-air mixing characteristics. Moreover, this increased fuel momentum can potentially alter the flow field in both, the premixing section and combustion chamber.

$$J = \frac{\rho_{fuel} U_{fuel}^2}{\rho_{air} U_{air}^2} \quad (2.22)$$

MAIN CONCLUSIONS RESEARCH REICHEL

The main conclusions and findings of Reichel's work are summarised below:

- Fuel reactivity, particularly hydrogen content, affects combustion stability limits in particular flashback propensity.
- Different measures against flashback, such as axial injection and fuel momentum, can interact and extend flashback limits effectively.
- Central non-swirling air injection on the central axis extended stability/flashback limits without compromising NOx emissions over the investigated setpoints.
- Fuel-air momentum ratio (J) is identified as a key parameter for extending flashback limits.
- Suitable estimators for flashback resistance include the axial location of VB for isothermal conditions and high values of upstream flame front location (x_f) for reacting conditions.

2.4.3. DUAL SWIRLER CH₄/H₂ STABILISATION REGIMES (IMFT)

Recently the Institute of Fluid Mechanics in Toulouse, which is associated with the university of Toulouse, have published several papers regarding their work with hydrogen and methane in their experimental lab as well as the accompanying LES analyses. Most of the papers, regarding experimental work, that were published in recent times were written by Marragou *et al.* [118, 141–143] and accompanied by numerical analyses of Aniello *et al.* [119]. Their combustion system called HY-LON (HYdrogen LOW Nox), which is introduced to as MIRANDA DFDS in older papers, is a newer version of their well known MIRANDA system. It is composed of a Dual Fuel Dual Swirlers system which sees the premixed methane being injected axially through a dedicated swirler ($S_{we} = 0.67$) around the center line while the hydrogen is also injected axially but separately through a different channel with its own variable swirler ($S_{wi} = 0 - 0.9$) which is on the center line as well. A cross-sectional view of the injector and chamber are given in Fig. 2.49. Noteworthy is the fact that both mixtures are not released in the chamber at the same axial length but the central hydrogen pipe ends earlier to allow for mixing of the flows before the complete mixture is dumped in the chamber. This difference between the end points of both tubes is called the recess length and can be changed, in fact the effect of changes in these parameters is the subject of one of the published papers [143]. This DFDS system is similar to the AAI employed by Reichel [5] and van den Bergh [47] discussed in the previous sections apart from three key differences. One being that hydrogen instead of air is flowing through the inner injector, the other that this stream can be swirled. The third one is the absence of a premixing tube.

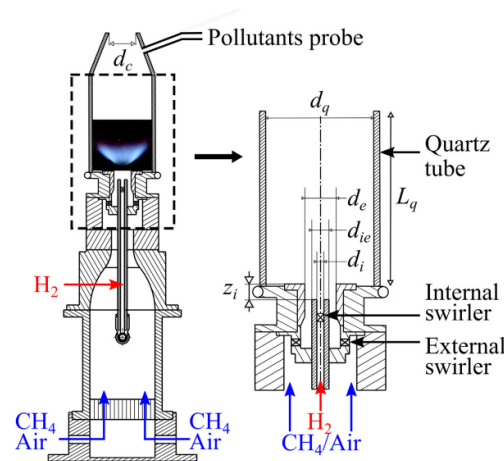


Figure 2.49: HY-LON experimental set-up [142]

Quite an extensive range of experiments have been performed varying the total mass flow, equivalence ratio and hydrogen content to name a few. These flames have also been inspected with several different measuring techniques: pyrometer, chemiluminescence, PIV, Raman scattering and emissions. Noteworthy, are the experiments with no inner swirl, visualised in Fig. 2.50, for increasing hydrogen contents as these are similar in nature with the set-up at the TU Delft apart from the central air injection being done with hydrogen.

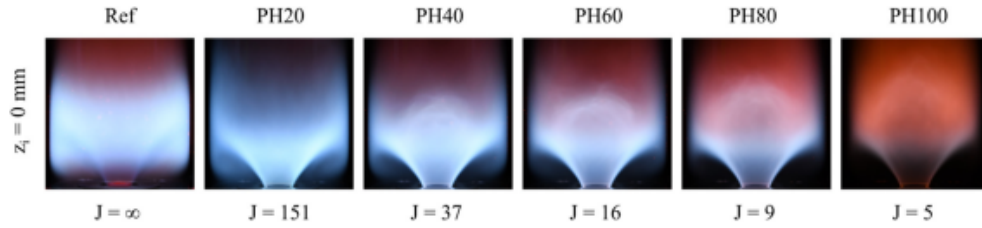


Figure 2.50: Reference chemiluminescence images of flames with various hydrogen contents with no inner swirl [142]

MAIN FINDINGS IMFT

The main findings of the various papers published by the research center are listed below:

- Central non-swirling hydrogen injection on the central axis extended stability limits without compromising NOx emissions.
- Fuel-air momentum ratio (J) is identified as a key parameter for extending flashback limits.
- Swirling the internal hydrogen channel is necessary but not sufficient to lift hydrogen/air flames.
- Among the injector geometries studied, hydrogen injection velocity predominantly controls flame stabilization, while air velocity has a weaker impact on flame shape transitions.
- The apex of the V-shaped flame narrows and sharpens as hydrogen content increases at a fixed thermal power.
- Increasing equivalence ratio or hydrogen bulk injection velocity helps lift the flame, with the hydrogen lift-off velocity increasing with hydrogen combustion power.
- At constant thermal power, CO emissions decrease, but NOx emissions increase with hydrogen content. NOx emissions are correlated with the stabilization mode.
- Partially premixed flames anchored to the central fuel lance produce higher NOx levels, which decrease significantly when the flame is lifted.

2.4.4. LEGACY RESEARCH AT TU DELFT COMBUSTION LAB

The swirl stabilized combustion research with AAI at the TU Delft started back in 2021 with the numerical and cold flow experiments of an injector by Vermeijlen [144]. His work was followed by the development, construction and commissioning of the swirl combustor set-up by van den Bergh [47]. The thesis marked a milestone for the lab and enabled the research of combusting mixtures with the swirl stabilised AAI injector it is further discussed in Subsec. 2.4.4.1. The experimental research is currently lead by Link *et al.* [145]. Next to the experimental research there is a numerical research counterpart for the research lead by Ferrante *et al.* [104] as presented in Subsec. 2.4.4.2. Parallel to the numerical work there is the experimental work by Link *et al.* [145].

DEVELOPMENT OF THE SWIRL COMBUSTOR BY A. VAN DEN BERGH

In the master thesis by van den Bergh [47], the original design of the combustor and the injector with the different swirlers is presented. The thesis project was heavily inspired by the design and earlier work by Reichel presented in the previous section. Like the thesis project for which this literature study is written it is also part of the same APPU project at TU Delft, which was introduced earlier in Sec. 2.2. This original swirl stabilised combustor design developed by van den Bergh in 2016 is currently located at the TU Delft combustion lab and will be the main test bed for the research and experiments that are envisioned for this thesis. The master thesis goes into great depth detailing the design as well as elucidating the entire design process and design choices for the experimental combustor. A 3D CAD model of the combustion chamber and a close up of the injector are presented in Fig. 2.51.

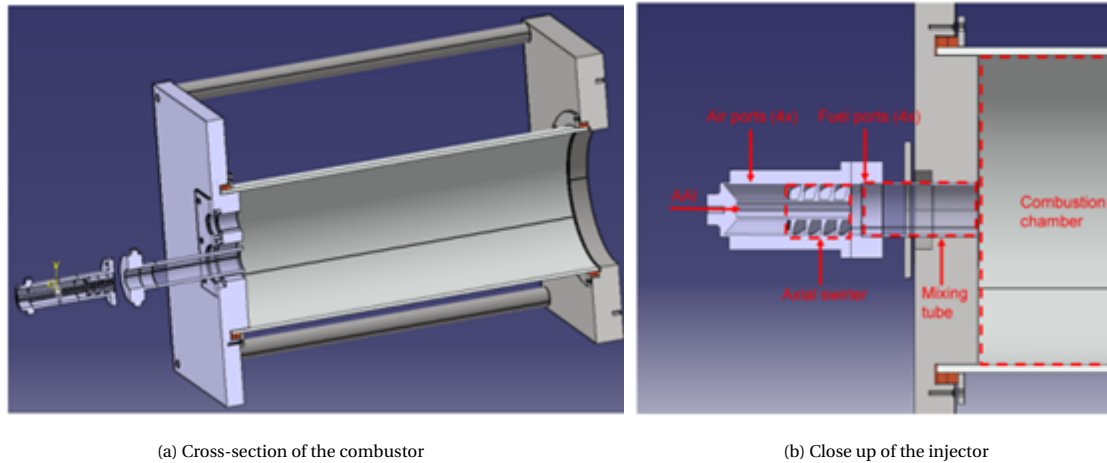


Figure 2.51: Some snapshots of the CAD renders of the combustor [47]

The development of the combustor address several important aspects with respect to behaviour of the set-up and its characteristics for which plenty of tests have been conducted and calculations have been made. As the same set-up will be used it is important to have a clear view on what exactly has been done to avoid unnecessary and double work and to identify what still needs to be done. One of the important calculations that has been done by [van den Bergh](#) is the relation between swirl intensity and the effective swirl number when AAI is increased. These have been visualised in Fig. 2.52 and reveal a strong drop in swirl intensity around the 10 % mark.

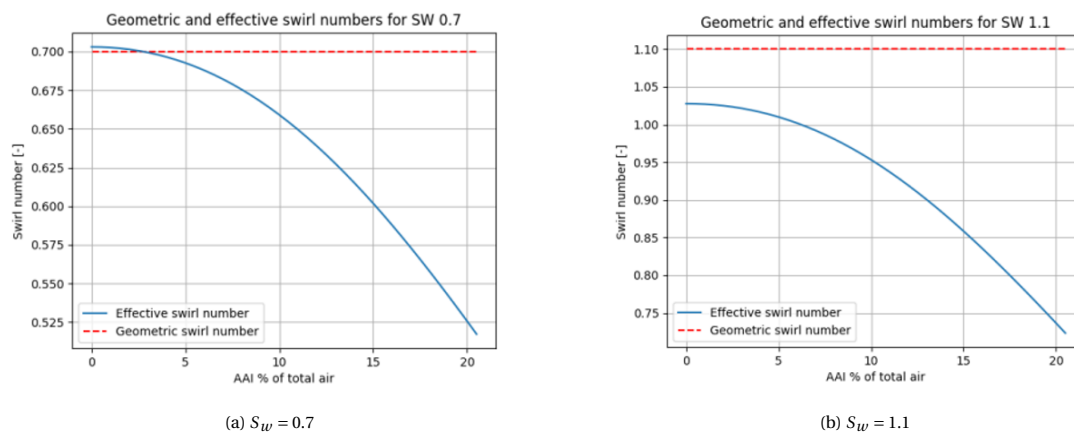


Figure 2.52: Theoretical swirl decay for different levels of AAI for different initial swirl intensities [47]

Next to designing and the one paper calculations [van den Bergh](#) also conducted several small test campaigns to commission the burner and gain some preliminary insights in the behaviour of the set-up. The commissioning of the burner included non-reacting and reacting flow experiments for different % of AAI, as shown in Fig. 2.53, but at a constant power and hydrogen enrichment level. The axial velocity profiles of the set points were visualized using a 2D2C PIV set-up, of which some results are shared below. He also conducted CH^* chemiluminescence experiments for both swirl intensities at different power levels.

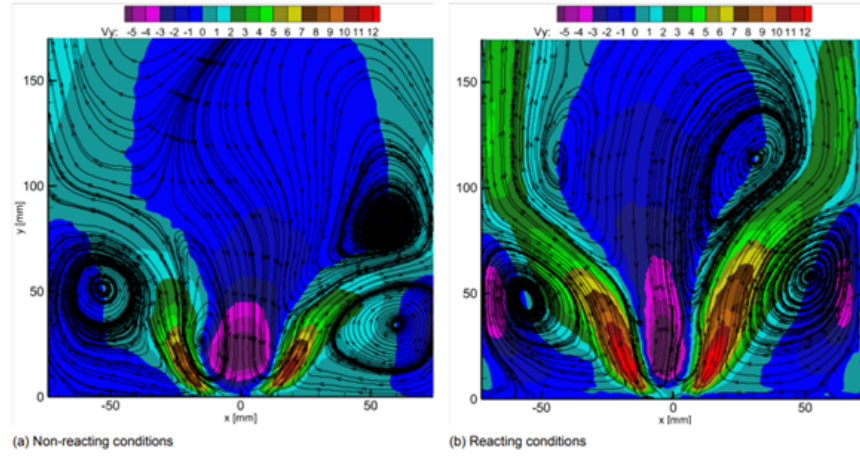


Figure 2.53: PIV results of both non-reacting and reacting flows for $S_w = 0.7$ and 5% AAI by [47]

Main Take-aways of Combustor Development and Commissioning

The main conclusions of his thesis are formulated below:

- When comparing non-reacting and reacting flowfields for a given SW (swirling) and AAI percentage, the effect of the interaction between combustion and swirling flows becomes evident. This effect is notably demonstrated by the presence of extra negative circumferential vorticity in the difference vorticity fields. It is essential to note that a precondition for this phenomenon to occur is that the non-reacting vorticity fields, under the specified S_w and AAI%, should already possess negative circumferential vorticity in close proximity to the center line. As a result, this condition leads to an additional reduction in axial velocity on the center line when compared to non-reacting conditions.
- There has not been an observable shift of the axial location of VB in reacting conditions compared to its non-reacting equivalent, for a particular S_w and AAI percentage.
- When comparing reacting conditions to non-reacting conditions, there has not been an observable extra axial velocity deficit close to the mixing tube. However, it is worth noting that the axial velocity is reduced more in reacting conditions further downstream. This observation aligns with the previously mentioned presence of extra negative circumferential vorticity in these regions. As a result, it can be concluded that the introduction of heat alters the flowfield, leading to a stronger CRZ, but it does not necessarily increase the flashback propensity of such flowfields.
- The percentage of AAI influences the flow field similarly under both reacting and non-reacting conditions, resulting in increased axial velocity near the center line close to the mixing tube outlet. This effect is more pronounced at high AAI levels and is moderated under reacting conditions, showcasing the impact of AAI on the flowfields.
- The recirculated mass flow distribution between the inner and outer zones indicates that in isothermal conditions, a significant portion of the mass flow recirculates in the ORZ. To address this, reducing the expansion ratio in future experiments is recommended to promote greater recirculation through the CRZ.

The experimental campaigns conducted by [van den Bergh](#) were relatively constrained in both scope and measurement techniques, primarily utilizing solely 2D2C PIV, chemiluminescence and a limited set of test points. Subsequently, the responsibility for advancing and broadening the experimental work using this setup was assumed by [Link et al.](#) as part of her doctoral research, which will be elaborated upon in the subsequent subsection.

NUMERICAL LES SIMULATIONS OF COMBUSTOR BY G. FERRANTE AND EXPERIMENTAL WORK BY S. LINK

A Large Eddy Simulation (LES) of the set-up [47] that will be used for the thesis experiments was conducted by Ferrante *et al.* [104]. More precisely he simulated the internal flow field of the lean premixed swirl-stabilized combustor with axial air injection at both non-reacting and reacting conditions, for methane and methane-hydrogen fuel mixtures using thickened flame combustion model (TFM) with detailed chemical kinetic mechanisms. The LES used an adaptive mesh strategy to adequately map the flame and the boundary layer regions. The results were validated for the methane flame against the experimental PIV measurements. The study saw the evaluation of the cases, A and B, consisting of a methane-only case (A) and one consisting of a methane/hydrogen blend (Case B), where the percentage of hydrogen amounts to 60% in volume. Both cases were at atmospheric conditions and were set at a power of 11 kW and 5% AAI.

Results from LES

Some of the results from the LES simulations comparing both cases are presented below in Fig. 2.54. Although, flashback is not observed in the LES simulations it is stated that there is some concern that the upstream propagation of the flame in the hydrogen blend could trigger it. Moreover, it is stated that the mixedness level of the mixture at the center line plays a critical role in enabling flashback. Since the power level is kept constant the hydrogen case operates at much leaner conditions explaining the lower temperature.

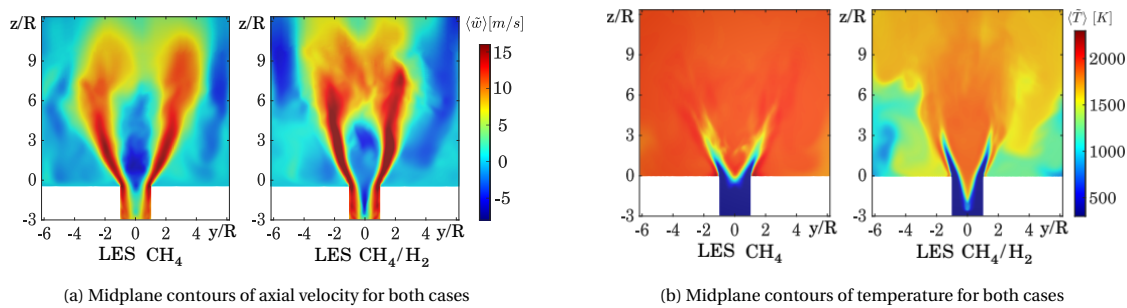


Figure 2.54: Some midplane contour plots from the numerical by Ferrante *et al.*[104]

A POD analysis of the results, of which a snippet is presented in Fig. 2.55, revealed the presence of a precessing vortex core (PVC) in both reacting and non-reacting conditions which is in agreement with the methane experiments conducted by Link *et al.* [145]. The PVC is noted to still be present in the LES simulation results when hydrogen is added, but the associated modes appear to be suppressed near the combustor entrance. Furthermore, Ferrante *et al.* [104] stresses that PVC causes variations in equivalence ratio which could couple with heat release and the vorticity generated by the recirculation bubble to trigger combustion induced vortex breakdown, and thus accommodate a flashback event.

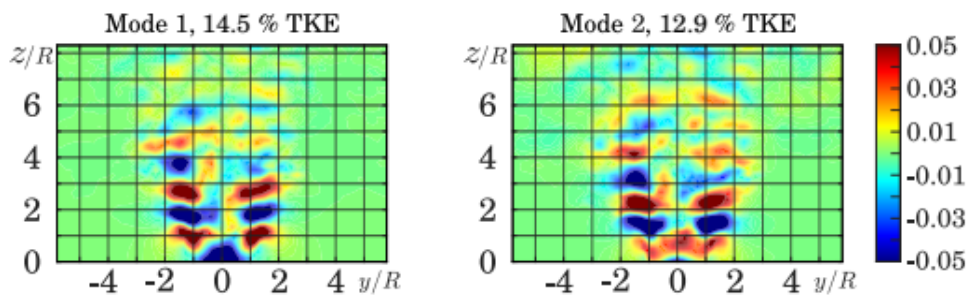


Figure 2.55: First two POD modes computed from LES for the non-reacting, methane-only case [104]

Regarding the analysed emissions the simulations showed that both the CO and NO sharply decrease when hydrogen is added to the mixture. The reduction in CO is attributed to the lower concentration of methane and the NO reduction can be attributed to the lower flame temperature due to the leaner operating conditions. However, the limited scope of the study, i.e. just the two cases, is unable to explain several findings

by S. Link in her experimental PhD research like for example the increase in NO when hydrogen content is increased even further even though the flame is leaner and should therefore be colder.

2.5. EQUIVALENCE RATIO OSCILLATIONS

In previous chapters technical challenges associated to hydrogen combustion and topics like flashback and flame blowout have been introduced. There is one more major technical challenge associated with lean pre-mixed systems, namely that of combustion instabilities. Lean premixed systems are more susceptible to combustion instabilities, more specifically thermoacoustic instabilities, because the flame properties like the burning velocity are more susceptible to perturbations in equivalence ratio. Given that state-of-the-art gas turbines operate at 10–50 bar oscillations may reach 1–2% of the static pressure, which is accompanied by severe structural damage, higher emissions, and a stark drop in efficiency. Therefore, combustion instabilities will substantially limit the operational range of the engine. [101, 146]

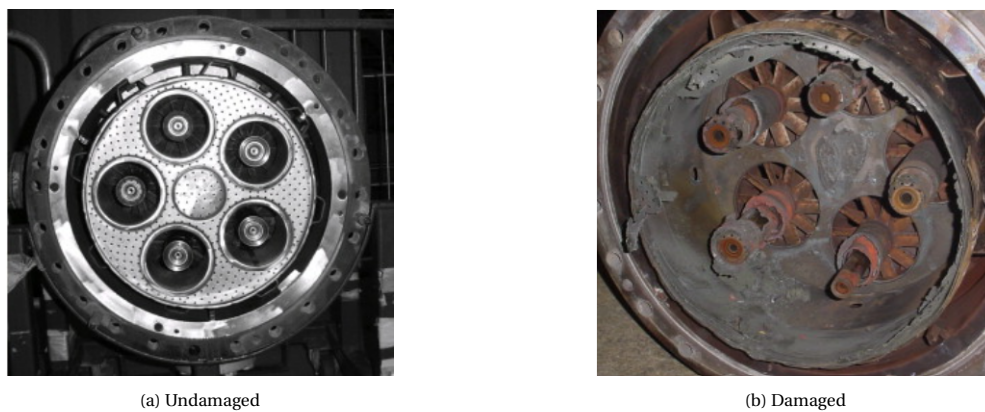


Figure 2.56: Combustor manifold before and after sustaining damage due to combustion instabilities [147]

Combustion instabilities in general are a critical phenomena observed in engines and gas turbines, characterized by unsteady behavior of combustion processes within their chambers. These instability results in fluctuations in key parameters, such as pressure, entropy, temperature, and flame characteristics. It manifests as pressure oscillations, temperature variations, and flickering flames, accompanied by strong acoustic waves and an increase in mechanical stress as a result of resonance, local hot spots, cooling regions, etc. . These combustion instabilities can have detrimental effects on system performance, emissions, and mechanical integrity. [69]

There are several unsteady flow and flame processes that can lay at the basis of a combustion instability as shown in Fig. 2.57. Furthermore, a distinction based on frequency can also be made to categorize them into three groups. High-frequency instabilities (above 1000 Hz) are caused by the interaction between acoustic disturbance and flame stretching. Intermediate-frequency instabilities or a buzz occur between 250 Hz and 1000 Hz. Lastly low-frequency instabilities⁸ (below 250 Hz) or cold tones increase in amplitude when temperature decreases. Sometimes when these low-frequency instabilities originate from a coupling of the combustion chamber with the feeding system they are referred to as chugging. [101, 148]

⁸Indirect and composition noise also appears at low frequencies

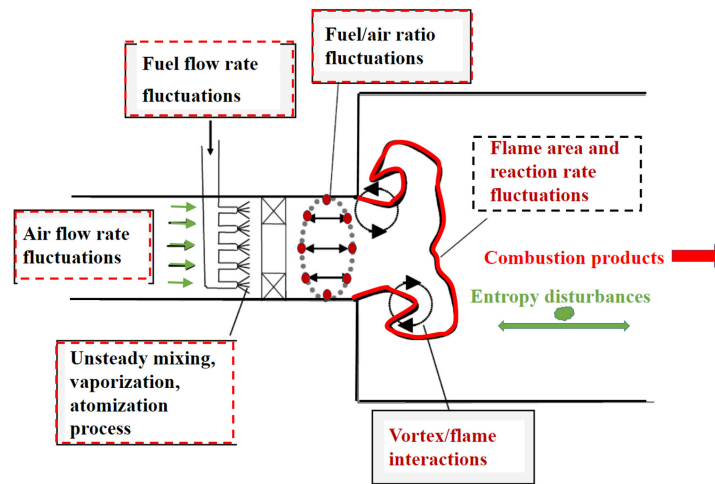


Figure 2.57: Unsteady combustion instability sources in gas turbine engines [149]

2.5.1. THERMOACOUSTIC INSTABILITIES

Thermo-acoustic instabilities specifically stem from the interaction of the unsteady heat release by the flame being in phase with the pressure oscillations which adds energy to the perturbation field. There are multiple origins that contributed to these variations but in premixed systems the most significant mechanisms are fuel/air ratio oscillations and vortex shedding. Both trigger a feedback loop, visualised in Fig. 2.58, where the pressure fluctuations in the combustor cause pressure and velocity fluctuations in the nozzle which translate to perturbations in the fuel flow rate and/or the vorticity. These fluctuations are naturally convected to the flame where they result in fluctuations in the heat release, which develops the pressure oscillations and the loop is closed. Note that this only happens when the Rayleigh criterion is met, i.e. combustion instabilities are excited only if the oscillation of sound-pressure and a flame's heat release rate are in-phase and the frequencies match [69, 148].

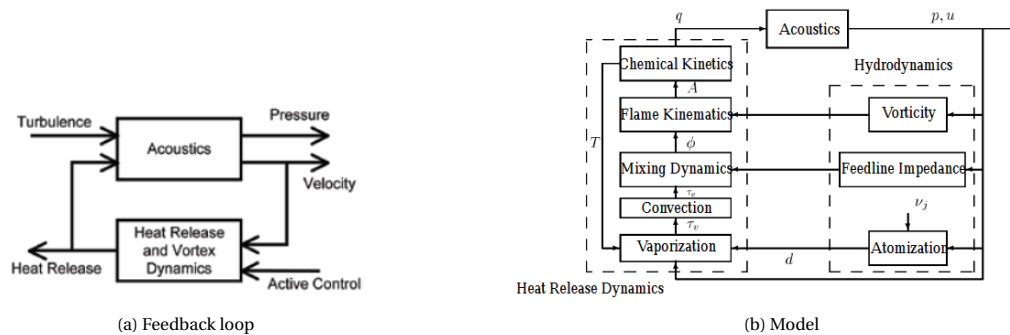


Figure 2.58: Combustion instability feedback loop [150] and model [151]

Fuel/air ratio oscillations can be attributed to acoustic oscillations in the premixer section, which alters the fuel and/or air supply rate. This can produce a reactive mixture with a periodically varying equivalence ratio. These mixture fluctuations are convected to the flame where it translates to heat release fluctuations. This coupling between the premixer acoustic and the fuel system is also affected by the pressure drop over the injector which varies with the fuel volumetric heating value. [69, 101]

The vortex shedding mechanisms involves large scale, coherent vortical structures that stem from flow separation from flame holders and rapid expansions, shear layer roll-up, and vortex breakdown in swirling flows. These vortices will disrupt the flame front when they come in contact with it. This will result in oscillations in the heat release rate. [69, 101]

COMBUSTION INSTABILITY CONTROL

Traditionally, combustion instability control is achieved passively and looks to breakdown/prevent the feedback loop by breaking the symmetry of the chamber or adding in vibration dampeners or fuel staging. [150]

For example, in 2016 Williams *et al.* [152] effectively suppressed thermoacoustic instabilities at various pressures in a swirl stabilised with a porous insert. It is also effective in reducing combustion noise over the entire frequency range with respect to different operating pressures, equivalence ratios, and reactant flow rates. Generally, passive methods can be broken down in the following categories:[153, 154]

- **Damping elements and silencers:** Turn acoustic energy into heat. They simply reduce the amplitude of the waves at the cost of additional pressure losses.
- **Helmholtz and lambda/4 resonators:** Disrupt the evolution of combustion instabilities by disturbing the resonant sound fields. Only effective for a limited range of frequencies. They can also be rather large for lower frequencies.
- **Baffles:** Prevent any resonant sound wave from forming by disrupting the sound waves. Can burn-off and even negatively impact the combustion process itself. Also incur high pressure losses.
- **Detuning:** The geometrical aspects of the combustor can be changed to alter its eigenfrequency. Quite complicated to achieve and the effectiveness frequently limited to specific frequencies. The changed geometric and material aspects of the chamber will also alter the heat loss to the combustor walls which in turn will affect the combustion process itself.

Active Instability Control

Significant progress was made with passive designs, but achieving the next level of performance improvement and emissions reduction requires integrating active control in the combustor. [150] Moreover, in cases where fuel composition can vary widely, such as multi-fuel systems it becomes harder and harder to develop a static design that is stable over all conditions and for all potential fuels. Other than restricting certain operating regimes depending upon fuel composition, active instability control might help broaden this range. [155]

In active control an incipient instability is dynamically detected and corrected by swiftly increasing or decreasing the fuel flow, or other modulation parameter, and prevent the instability from happening. Naturally, this requires online models of the heat release and acoustic phenomena that can produce optimized actuation signals based on measurements. The signals are then send to a (high frequency) mass flow controller or loudspeaker that will change the equivalence ratio through flow modulation or pressure out of phase waves. However, the quick response time can prove difficult in practice [150].

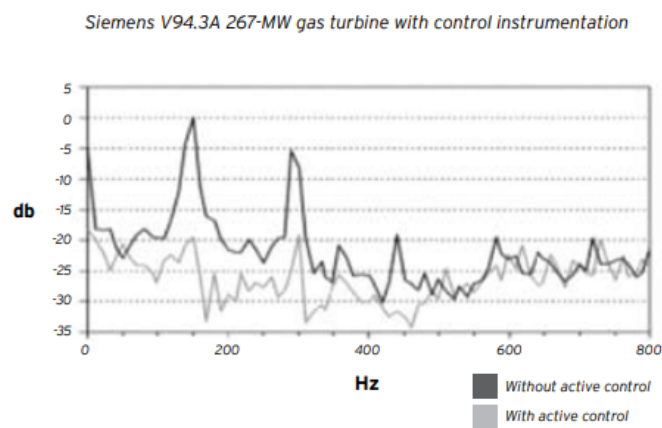


Figure 2.59: The reduction in acoustic pressure versus acoustic frequency [156]

Many successful lab-scale demonstrations of active instability control have been reported in literature. A historical review of the technique was conducted by Culick and Palm [157] and more recently by Zhao *et al.* [149]. These reviews highlighted two mainstream approaches: either that the offset for the instability is created through air or fuel flow modulation with a valve [156] or by loudspeaker that generates an out of phase pressure wave[158]. However, the review by Culick and Palm [157] concludes that the research into active instability control has slowed down as cheaper passive solutions have been found to sufficiently dampen the

instabilities. Consequently, the experimental research into active instability control has slowed down. With most of the academic research being on a laboratory scale since the work of [Hermann and Orthmann](#) [156] on an a Siemens heavy-duty industrial gas turbine. Most current research into active instability control is done with CFD. [159] Yet there remain several reasons to still research active instability control: it has a smaller footprint regarding weight and size compared to passive systems, can be retrofitted more easily and offers more flexibility in adapting to different combustion conditions.[160] This does not mean that the research into related topics has slowed down. For example, recent papers by [Kather et al.](#) [146], [Deng et al.](#) [161] and [Kwak et al.](#) [162] employ a loud speaker to study the effect of acoustically forced flames. The use of a high frequency valve to modulate the fuel flow to study the effect of fuel-feeding excitations has been applied in even a more recent paper by [Hashim et al.](#) [101].

In industry the most prominent developer of active instability control is a German company called IfTA Systems [160] that offers an Active Instability Control system for a large market share of gas turbines like the Siemens Energy standard of gas turbine types 4000F, 8000H and 9000HL. They offer a system of sensors, actuators and control system architecture depending on the encountered frequencies and system specifications. The company is founded on the scientific work of [Hermann and Orthmann](#) [156] which will be mainly covered in the next section. Next to IfTA, GE offers an autonomous tuning software for its aeroderivative gas turbines that optimises the operating point based on emission, acoustic and load limitations. This system is technically not on par with the high frequency capabilities of the IfTA systems it does offer some active control capabilities in fact it is controlled by AI and machine learning. [163] It should also be noted that the RQL designs upon which the current GE aeroderivative gas turbines are build suffer far less from combustion instabilities than lean premixed systems. [157]

2.5.2. EFFECT OF EQUIVALENCE RATIO FLUCTUATIONS

In this section the various areas of research that are relevant for the proposed thesis subject are covered. Research pertaining to fuel excitation through valves is presented in [Subsec. 2.5.2.1](#). The effect of hydrogen enrichment in combination with fuel system fluctuations is covered in [Subsec. 2.5.2.2](#). Lastly, the effects of a premixing tube on equivalence ratio fluctuations is covered in [Subsec. 2.5.2.3](#).

VALVE INDUCED FUEL-FEEDING EXCITATIONS

Laboratory Scale

Through the years there have been several studies into various strategies that adjust the mixing process inside the combustor. A more advanced technique known as Fuel feeding excitation sees the amount of fuel being varied by a number of different control methods, such as fuel metering valves, fuel injectors, and control algorithms in order to maintain a stable flame. The most recent study in the field was conducted by [Hashim et al.](#) [101] and utilises a proportional directional control valve (PCDV) to study the effect of the excitations on the turbulent swirling flow structure in their model lean gas turbine combustor. Earlier research showed that stability modes inside the model combustor were more responsive when their frequencies harmonized with each other and that a stabilisation case is possible with when the fuel frequency matched the combustion chamber pressure wave fluctuations [164]. The original link between large variations in pressure in the inner recirculation zone and shear-layer instabilities was observed by [Paschereit et al.](#) [165]. This was expanded upon by [Külsheimer and Büchner](#) [166] who showed that the self-excited frequency ($f_{natural}$) controlled coherent ring-vortex structure formation. Furthermore, it was shown that the fuel-feeding excitation affected the swirling flow structure and flame stabilization due to the periodic fluctuation of velocity components and the appearance of the PVC [167, 168], albeit it being suppressed more in reacting conditions [169]. The instabilities increased when the PVC spiraled more around the central axis [170, 171]. In case of high-frequency fuel modulation a mixing process near the fuel injector takes place, resulting in the amplification of equivalence ratio fluctuations [164, 172].

Most of the current day research relies on Large-eddy simulation (LES) to effectively capture unsteady features in both non-reacting and reacting systems in gas turbine combustors [101, 173, 174]. The aforementioned research of [Hashim et al.](#) [101] employs both experimental and numerical research to study the effect of the induced equivalence ration fluctuations.

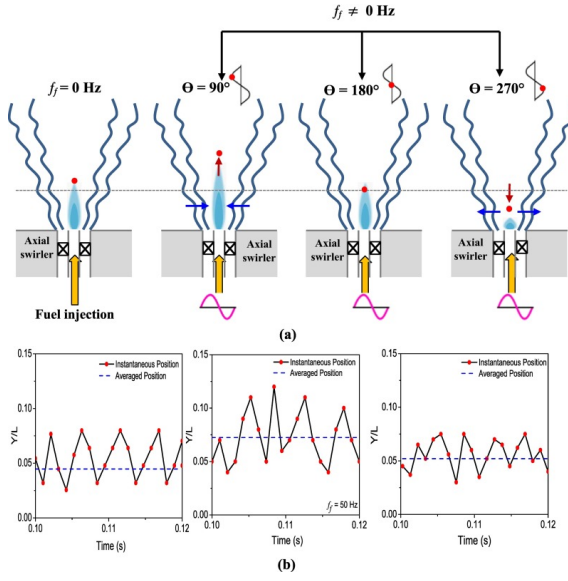


Figure 2.60: The effect of fuel oscillations on the stagnation point[101]

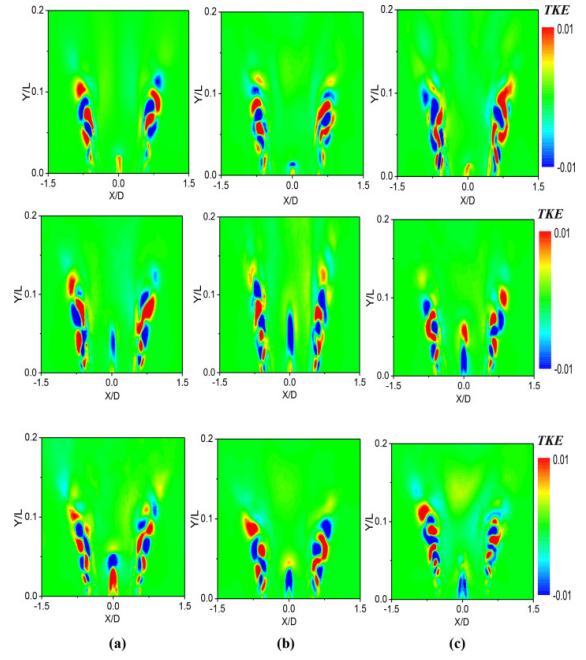


Figure 2.61: The POD modes on x-y plane-based turbulent kinetic energy (TKE) for the (a) 1st mode, (b) 2nd mode and (c) 3rd mode at $f_f = 0$ Hz (1st row), $f_f = 50$ Hz (2nd row), and $f_f = 200$ Hz (3rd row).[101]

In general, the flame that was observed by Hashim *et al.* [101] was characterised by a bubble vortex breakdown in all cases researched with forced fuel fluctuations. Moreover, the enhanced mixing originating from the excitations facilitated better combustion performance and hence reduced the extension of the central recirculation zone and the flow instability near the combustor wall. The effect that the equivalence ratio fluctuations can have on the flame, more precisely the stagnation point, for different phase angles (θ) can be seen in Fig. 2.60. These fuel-feeding excitation causes the upstream stagnation point to undergo nonlinear oscillations in the radial direction, moving it closer to the central axis of the model combustor where the frequency of these oscillations is influenced by the characteristics of the fuel-feeding excitation itself. This resulted in more symmetrical vortices being witnessed in the flame structure meaning that the fuel excitations promote more stable and organized flow patterns within the gas turbine combustor. Furthermore, the fluctuations accelerate the regions of concentrated vorticity at the interface between the fuel and air streams creating stronger small vortices. This will create co-winding counter rotating helical vortices, which are located at the beginning of the reverse flow region and the shear layer zone, as evidenced by the POD analysis presented in Fig. 2.61.

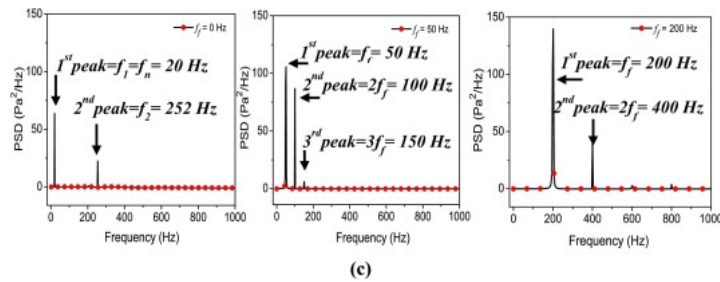


Figure 2.62: FFT of the effect of fuel oscillations on the stagnation point[101]

Industrial Scale

As alluded to in the previous section the active instability control offered by IfTA [160] is centered around the work of Hermann and Orthmann [156]. Consequently, his work presents the only available source in academic literature on fuel flow modulation for large scale industrial gas turbines to the author's knowledge

[159]. They fitted the burners on type Vx4.3A land-based Siemens gas turbines with a direct drive valve that allowed for mass flow modulations within the pilot gas supply of the burner. These modulations are anti-cyclical to the oscillations characterising the heat release rate within the flame, thus extinguishing them. They looked into three configurations: A) active control at switch-over from diffusion flame to premixed flame and B) active control at load change during premixed operation C) active control at start-up and at lower part load operation of the gas turbine. It is mainly the second option that is relevant for this work.

In a normal configuration the engine experiences increased amplitudes for the third harmonic (270 Hz) as early as at 60% of base load. At approx. 80% of base load, this frequency shifted to 170 Hz, corresponding to the second harmonic of the combustion chamber. The active instability control system is able to dampen the amplitudes of the harmonic responses by 35 – 60% depending on the harmonic frequency. This is achieved by introducing equivalence ratio fluctuations that are anti-cyclical to the oscillations characterising the heat release rate.

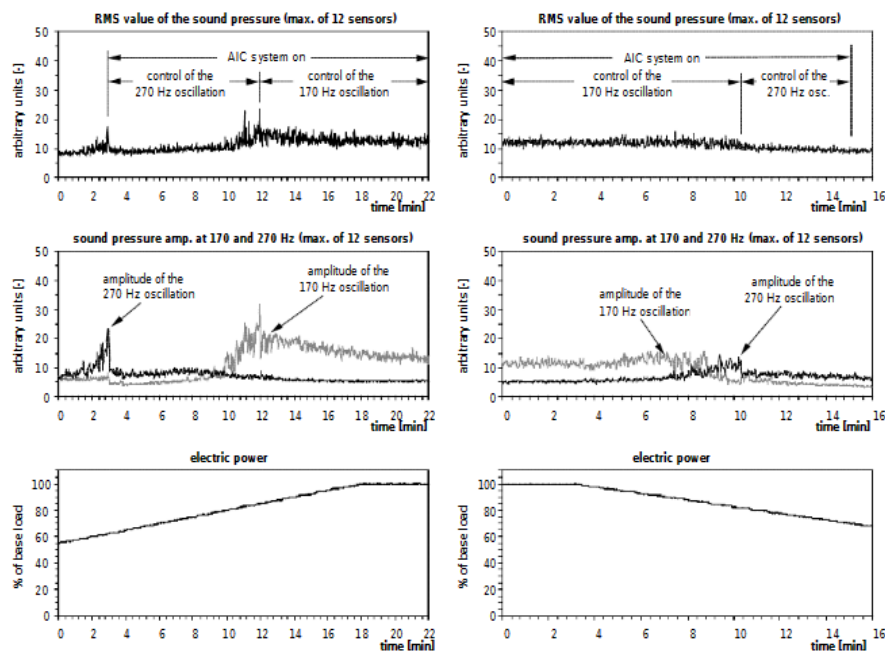


Figure 2.63: Effect of active instability control visualised in sound maps [156]

In contrast to passive measures, AIC systems offer several advantages. Unlike the limited frequency or operating range damping effect often associated with passive measures in combustion systems, the AIC stands out for its remarkable flexibility and extensive operational range. It was showed that the AIC and thus equivalence ratio oscillations can effectively be tuned to different frequencies to counter thermoacoustic instabilities. Furthermore, AIC has the potential to enable the operation of specific combustion systems with reduced emission levels. In the aforementioned premixed mode tests, activating the AIC made it feasible to operate the gas turbine with lower pilot gas mass flows compared to when the system was not in use. Given the significance of pilot flames in influencing NO_x emissions in gas turbines, the AIC system facilitated a reduction of more than 60% in certain cases.

EFFECT OF HYDROGEN ON EQUIVALENCE RATIO FLUCTUATIONS

Deng *et al.* [161] researched the effect of equivalence ratio fluctuations induced by acoustic excitations (90–240 Hz) in hydrogen–methane lean premixed swirling flames. Whereas, both Joo *et al.* [172, 175] and Kwak *et al.* [162] used a siren to induce fuel velocity oscillations (0–600 Hz). The former looked into H_2/CH_4 syngas flame dynamics in a partially premixed model gas turbine combustor (swirl-stabilised), the latter looked into forced response of H_2/CH_4 flames in a dual-nozzle swirl-stabilized combustor. The clear effect of hydrogen enrichment in combination with the oscillations is visualised in Fig. 2.64 and Fig. 2.65 for each respective study.

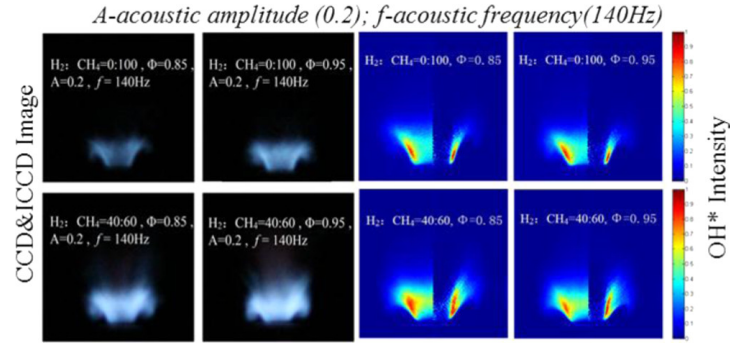


Figure 2.64: Effect of hydrogen addition on flame front [161]

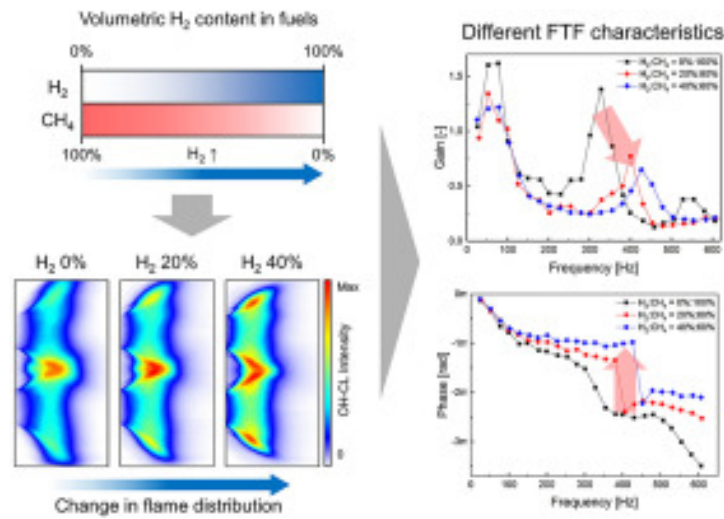


Figure 2.65: Effect of hydrogen addition on flame acoustic response[162]

Deng *et al.* [161] reported that the addition of hydrogen caused either an increase or decrease in the magnitude of the local maximum gain value depending on frequency and phase shift. Kwak *et al.* [162] on the other hand noted a decrease in local maximum gain values and a shift in frequency. It was also observed that the effect on the curvature of the flame was dependent on the forcing frequency. As the higher velocity disturbance more effectively intensified the compression of the flame length, which enhanced the mixing of the high-burning gas and the unburned gas. The latter also noted that hydrogen addition also caused a delay in the response of the flame to incident velocity fluctuations.

The work of Joo *et al.* [172, 175] put more emphasis on the actual influence of the fuel line itself. For example, it concluded that the frequency peaks were dominantly affected by the resonance frequency of the fuel line (classified as a quarter-wave frequency). When the flow was modulated with these peak frequencies, the periodic fluctuation of the OH^* intensity amplified the heat release fluctuation in the combustion zone. They also observed a linear trend between the phase shift of the flame and the fuel line perturbation and the Strouhal number as shown in Fig. 2.66. The Strouhal number relates the unsteady forces to the steady forces as shown in Eq. 2.23 where $f_{forcing}$ is the forcing frequency, L_f the characteristic flame length and $\bar{v}$ are the veloc-

ity fluctuations. Lastly, the measurements showed that the system became more robust and insensitive to perturbation with increasing hydrogen content.

$$St = \frac{f_{forcing} L_f}{\bar{v}} = f_{forcing} t_{conv} \quad (2.23)$$

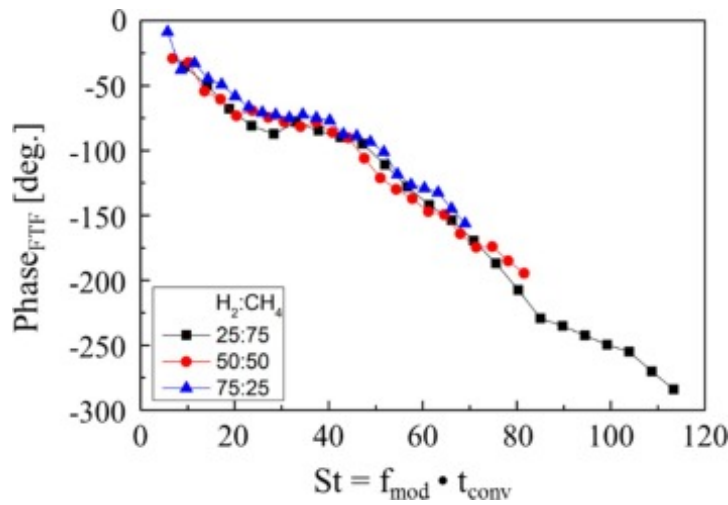


Figure 2.66: Relation Strouhal number and [175]

Hydrogen Injection for Combustion Instability Control

An interesting use of hydrogen in combination with combustion instabilities was also researched by [Barbosa et al. \[176\]](#). They used local hydrogen injection as a way to control (thermoacoustic) combustion instabilities in a premixed propane flame. They used several diagnostic techniques to look into the effects of the injection on combustion instabilities and emissions and characterise the system. They also evaluated the case where the central hydrogen injection jet was swirled.

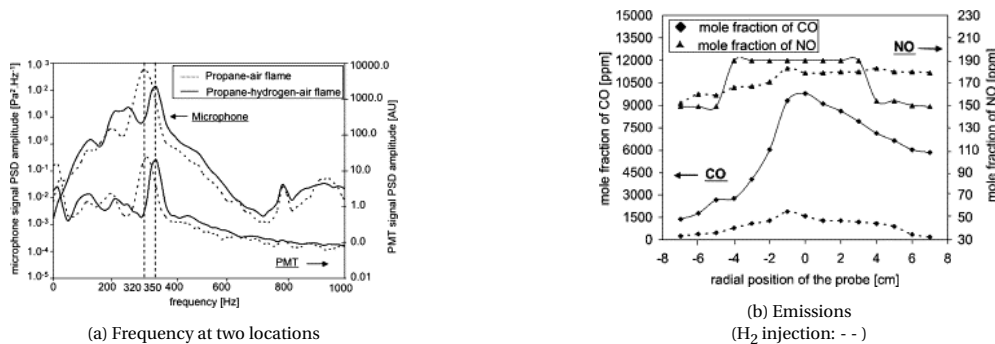


Figure 2.67: Effect of local hydrogen injection for combustion instability control [176]

The experiments showed promising control regarding the use of hydrogen injection as a combustion control technique. Especially, the swirled hydrogen injection showed to be more efficient in reducing the pressure oscillations. Diagnostics suggest that the hydrogen and hydrocarbon combustion zones are separated. However, a more detailed analysis regarding the interaction mechanisms and reaction paths would be needed to confirm this. Furthermore, emission measurements indicated that the hydrogen injection caused higher CO and NO emissions. This is attributed to the higher combustion temperature of hydrogen.⁹

⁹The power output probably also increased with the hydrogen injection

EFFECT OF PREMIXING TUBE ON EQUIVALENCE RATIO FLUCTUATIONS

Kather *et al.* [146] used the same set-up as Reichel [5], that was discussed in the previous chapter, to research the interaction of equivalence ratio fluctuations and flow fluctuations in acoustically forced swirl flames. Instead of a mass flow controller to induce the oscillations he used a loudspeaker to induce them. However, his research gives insight on how a premixing tube after the the swirler affects the induced fluctuations. The flow is measured with microphones before the swirler and with Tunable Diode Laser Absorption Spectroscopy (TDLAS) in the premixing tube as shown in Fig. 2.68. The flame behaviour itself was analysed using PIV and both OH^* and CH^* chemiluminescence.

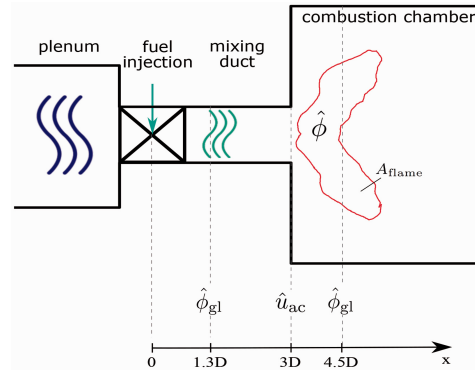


Figure 2.68: Schematic of the set-up with measuring locations [146]

Kather *et al.* [146] noted in his conclusion that the length of the premixing tube strongly influences the phase and gain of the induced equivalence fluctuations. As the model indicated that the effective diffusivity was much higher in the premixing tube than in the combustion chamber. Furthermore, his experimental measurements showed that the equivalence ratio fluctuations decayed with higher acoustic forcing amplitudes which is attributed to a non-linear saturation process at the fuel injector.

3

DISCUSSION & THESIS OUTLINE

From this assessment a research objective is established in [Sec. 3.2](#) from which the research question(s) naturally follow in [Subsec. 3.2.1](#). Lastly, the project planning including the Gantt chart are presented in [Sec. 3.3](#).

3.1. ASSESSMENT OF THE STATE-OF-THE-ART KNOWLEDGE

The current state of the art research and development of air and land based turbines focuses on reducing emissions and considers implementing lean swirl stabilised systems that operate on alternative fuels in order to reduce emissions. Hydrogen is seen as one of the front runners for these fuels given its promising role in the energy transition to come and its lack of a carbon atom. However, the supply chain of hydrogen still needs to develop, and hence a transition period is envisioned where multi-fuel systems will become the norm and existing systems are retrofitted to allow for enrichment. In addition, hydrogen enrichment increases the burning velocity and temperature of the fuel mixtures and influences the dynamics of the flame significantly. Therefore, a lot of research has been published recently with regards to hydrogen enrichment in various combustion systems.

A critical aspect of these lean systems is that they are more susceptible to combustion instabilities. These instabilities can stem from various sources and can trigger strong pressure waves that damage the structure while increasing emissions. Traditionally, these instabilities are suppressed using passive techniques like dampers, baffels and resonators or by redesigning the combustion chamber to alter its eigenfrequency. However, in multi-fuel systems this becomes harder as these passive methods only have a limited frequency range for which they are effective and the fuel composition plays an important role in determining the frequency of the instability. Active control on the other hand does not suffer from this limitation as the dampening frequency can be actively tuned and altered to match and dampen that of the combustion instability.

Active control is an instability control method that boasted a lot of interest and promise a few decades ago before it was mainly dropped in favour of cheaper and less complex passive solutions. Currently, there is some regained in the topic give the instability challenges lean multi-fuel systems pose. Active instability control sees the modulation of fuel flow and therefore equivalence ratio to combat combustion instabilities and break the feedback loop that sustains them. The modulation is achieved through actuators actively changing the mass flow or through acoustical and mechanical drives that generate oscillations throughout the fuel supply.

Very little research has been published that combines both hydrogen enrichment as well as fuel-excitation to combat combustion instabilities in lean, premixed, swirl stabilised combustors. This thesis envisions to lay the basis for the development and research into such system for the swirl combustor used at TU Delft sustainable aircraft propulsion lab by characterising the system with regards to these forced fuel-flow excitations.

3.2. RESEARCH OBJECTIVE

The thesis research objective is stated below:

"To (experimentally) investigate the effects on flame dynamics, stability and emissions of forced equivalence ratio fluctuations in a lean premixed axially swirl stabilized hydrogen enriched methane combustor through acoustic measurements, PIV, chemiluminescence and exhaust gas analysis respectively."

The focus of the thesis project lies in characterising the effect of low frequency, i.e. below 200 Hz, fuel line excitations in combination in H₂ enrichment in a swirl stabilized CH₄ flames with axial air injection. This is done on three levels, namely dynamics, stability and emissions. The project foresees the design and execution of experiments with regards to the aforementioned topics. The effects of the equivalence ratio oscillations will be analysed throughout the experiments by means of PIV, Chemiluminescence imaging, emission- and acoustic measurements. The purpose of the research is to characterise the effect of these equivalence ratio fluctuations on a hydrogen enriched system like the swirl burner with AAI. Furthermore, it will lay the ground work for a potential study and development of an active instability control system for a multi-fuel system.

3.2.1. RESEARCH QUESTION

In order to accomplish the objective that has just been formulated and clarified a research question and additional sub research questions are needed to guide and structure the research. The main research question is stated below.

How do forced equivalence ratio fluctuations affect the flame dynamics, stability and emissions of a stable axially air infused, hydrogen enriched methane flame in a lab scale technically premixed swirler combustor?

SUB-RESEARCH QUESTIONS

The main research question is elucidated by several subquestions to tackle the different facets of the research question. One can find them listed below in this subsection.

1. To what extend do the imposed equivalence ratio fluctuations alter the flow field of the enriched methane flame? And does the effect change with power level, valve frequency or hydrogen quantity?
2. How do the NO and CO₂ emissions of the hydrogen enriched methane flame change when equivalence ratio fluctuations are imposed on the flame?
3. To what extend does the imposed frequency of equivalence ratio fluctuations influence the stability, more specifically the acoustics and pressure waves, of the hydrogen enriched methane flame?
4. How do the findings related to the imposed equivalence ratio fluctuations on hydrogen enriched methane flames in a lab scale premixed swirler combustor translated to larger systems using a low order combustion instability model of the system?

3.3. PROJECT PLANNING AND GANTT CHART

The project is thesis project is divided into five phases and includes three test campaigns. The first phase pertains to this literature study and ends with the final submission of this report and the accompanying presentation being completed. This is followed by three test campaign phases; apart from the initial introductory campaign, these test campaign phases consist of a preparation phase, followed by the test week, and are ended with a data-processing and reporting phase. The goals of the different test campaigns will be discussed more in depth in the project plan and thesis report. The last phase is reserved for finalising the thesis report and ends with a public presentation, thesis defence, submission of the report, and archiving the obtained measurements/tools/documents. An overview is provided in [Fig. 3.1](#) with a larger version available in [Appendix A](#).



Figure 3.1: Planning thesis project

4

SWIRL-STABILISED COMBUSTOR SETUP

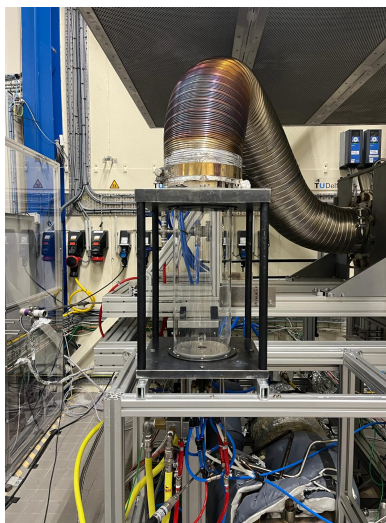
This chapter will present the swirl-stabilised combustor and broader lab facilities of the TU Delft sustainable aircraft propulsion lab. First the swirl-stabilised combustor and supporting infrastructure will be discussed in [Sec. 4.1](#) and [Sec. 4.2](#) respectively. The additional system necessary to inflict the equivalence fluctuations by modifying the airflow is introduced in [Sec. 4.3](#).

4.1. COMBUSTOR

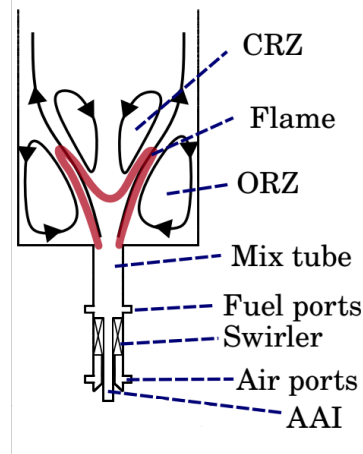
The combustor is made up of the swirl manifold, the combustion chamber, and exhaust duct. The other supporting systems like gas rack, control, ventilation are discussed in [Sec. 4.2](#). This section first gives an overview of the combustor and then zooms in on the manifold, more specifically the swirler. Finally, the combustion chamber and exhaust will be discussed.

4.1.1. OVERVIEW

The combustor setup used in the experiments is shown in [Fig. 4.1](#) together with a schematic that shows the main components as well as the main streamlines defining the Inner and Outer Recirculation Zones. The flame generally anchors along the red contour lines in the shear layer between these two zones. However, it may also anchor exclusively within the Inner or Central Shear Layer (ISL), or it might fail to attach to the injector inlet within the Outer Shear Layer (OSL). The general specifications of the combustor can be found in [Table 4.1](#).



(a) Lab



(b) Schematic [104]

Figure 4.1: Overview swirl-stabilised multi-fuel combustor TU Delft Sustainable Propulsion Lab

Table 4.1: Tu Delft Sustainable propulsion lab's lean premixed methane swirler

Lean (Technically) Pre-mixed Swirl Combustor			
Power	7–12 kW	Fuel	<i>methane, hydrogen</i>
Swirl number	1.1	Equivalence ratio	0.75 – 0.61
Temperature	1900–2200 K	Pressure	<i>atmospheric</i>
Air flow	167–250 nlpmp	H ₂ enrichment level	0 – 100%
Bulk velocity	7.8–10.7 nlpmp	Reynolds number	10,944 – 11,690%
Diameter	14.7 cm	Length	40 cm

4.1.2. SWIRL MANIFOLD

The swirl manifold, shown in Fig. 4.2, consists out of three components: the swirler with a central slit for the axial air to pass through, the fuel injector and the premixing tube. The main air enters the swirler through four ports into swirler where it is given a certain swirl depending on the installed swirler, then the fuel, i.e. methane and/or hydrogen, is injected radially and starts mixing in the premixing tube so that one gets achieves a technically premixed mixture before it flows into the combustion chamber and flame. The general dimensions are given in Fig. 4.3.

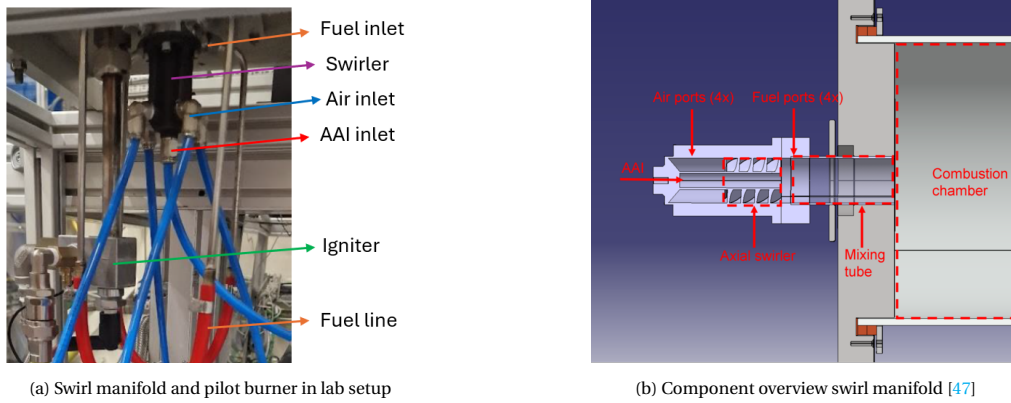


Figure 4.2: Overview swirl-stabilised multi-fuel combustor TU Delft Sustainable Propulsion Lab

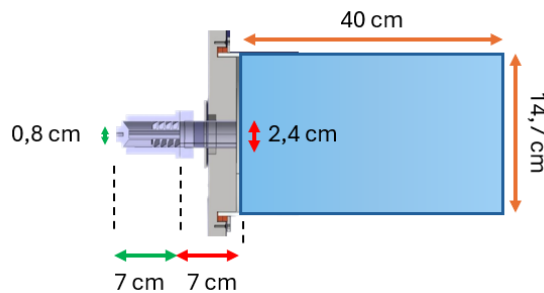


Figure 4.3: Dimensions Manifold and combustor

SWIRLER

The swirler is an axial swirler that gives the air flow the necessary swirl intensity to stabilise the flame as explained in Subsec. 2.2.2. The facility has two different variants of the same swirler shown in Fig. 4.4 that vary in the amount of swirl deflection they inflict on the flow. The swirl intensity of the different models varies between a swirl number of 0.7 or 1.1. The experiments mainly used the 1.1 model.

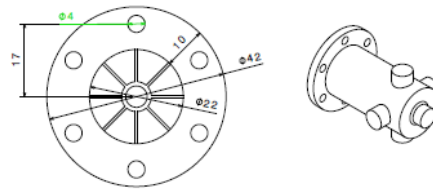


Figure 4.4: Swirler

AXIAL AIR INJECTION

To combat flashback it is possible to revert part of the main air away from the swirler to a central axial air injector passage that runs through the swirler along the center line. This will increase the axial velocity of the flow and increase the flashback resistance of the combustor. The amount of flow that is directed away is inputted as a percentage of total air flow by the operator in the control system of the swirl burner. The percentage of AAI is defined as $\chi = V_{Ax} / (V_{Ax} + V_{sw})$.

4.1.3. QUARTZ GLASS COMBUSTION CHAMBER

The combustion chamber houses the main flame and is interchangeable depending on the type of measurement one wants to take. Usually, the setup is equipped with Quartz glass variant to allow for easy visual access to the flame and to conduct laser or other optical measurements on the reaction zone. Quartz glass can withstand temperatures up to 1100°C and has good thermal shock resistance and thermal expansion coefficient characteristics. It has a length of 400 mm and an inner diameter of 147 mm and thickness of 3 mm.

4.1.4. COMBUSTOR EXHAUST

The combustion chamber is attached to the exhaust, as seen in Fig. 4.5, which safely removes the combustion product gases from the test section. The exhaust can be subdivided into three subsection: the flexible exhaust tube connecting the swirl combustor to the lab exhaust system, the cooling and thermocouples zone and the long rigid exhaust system that leaves the building. The flexible exhaust has an inner diameter of 102 mm and length of roughly 1.6 m. After the flexible tube section the exhaust gases are cooled before they enter the last section before they are released into the atmosphere.



Figure 4.5: Exhaust system

4.2. SYSTEM COMPONENTS

In this section, the supporting lab infrastructure that is essential for the operations of the swirl-stabilised combustor are discussed.

4.2.1. AIR SUPPLY SYSTEM

The air required for the combustion system is drawn from the dry air storage vessel that supplies the general air for all experiments for the entire High Speed Lab. The air the tank supplies is dry, particulate-free, and at a

nearly constant temperature thanks to its compressor, filtration system, and heat regenerator bed of ceramic pebbles. Exiting the tank, the air is then directed through a pressure conditioning system designed to lower its pressure to a usable range of 1-12 bars for laboratory operations. Each mass flow control unit includes a meter and a valve that adjusts according to the real-time flow, leading to minor fluctuations in air delivery to the combustor as the system strives to meet the targeted flow rate. The accuracy of these adjustments, as per the manufacturer's specifications, is within 1% of the full scale (FS). Within the lab, the air supply branches off into several lines, of which three are relevant: one directing the main air flow to the swirler, one directing the main airflow to the axial air injector, and the last one is used to cool the exhaust. Details and schematics of the air supply system are detailed in [Appendix B](#).

4.2.2. GAS RACK

The gas rack serves as the central hub for managing and regulating pressure for oxygen (O_2), hydrogen (H_2), methane (CH_4), and nitrogen (N_2) within the laboratory. It is equipped with four manual valves connected to the fuel and diluent lines, which must be activated to open the lines prior to initiating any experiments. These valves do not adjust the pressure levels but allow the flow of gases into the combustion system of the lab. Additionally, the gas rack is designed with a ventilation system to safely remove any potentially explosive or hazardous gases that might escape, ensuring a safe working environment in the event of a leak.

4.2.3. GAS SUPPLY SYSTEM

The external gas supply for the laboratory is situated in a secured area behind the building. This supply area accommodates four distinct groups of gas cylinders, which include hydrogen (H_2) and methane (CH_4) as fuels, along with nitrogen (N_2) and carbon dioxide (CO_2) serving as diluents. Each group comprises 16 B50 cylinders, each holding 33.3 L of gas at pressures varying between 1 and 180 bar. Pipelines extend from these cylinders into the lab, connecting directly to the gas rack. To initiate the pressurization and supply of fuel and diluents to the laboratory, the operator must manually open valves on the cylinders outside. Should there be an over pressure situation within the system, the gases are safely released into the atmosphere. Detailed information and layout of the gas supply system can be found in [Appendix C](#).

4.2.4. PILOT BURNER

A pilot burner is used to start-up the system and is used to ignite the main flows. It uses a spark plug to ignite the premixed air/methane pilot burner flows which can be specified by the user with regards to power and mixture ratio. The pilot flame is kept aflame until the system achieves a steady state, at which point the main flame becomes self-sustaining. The specific model used is the ZMI 16B200R, produced by Kromschroder, and is designed for atmospheric pressure operations with methane as the fuel source. It can be seen next to the swirler in [Fig. 4.2a](#). The rough power settings can be found in [Table 4.2](#)

Table 4.2: Typical ignition conditions for methane spark plug igniter

Typical ignition conditions			
Power	2.5 kW	Equivalence ratio	0.85

4.2.5. COMBUSTOR CONTROL AND DATA ACQUISITION SYSTEM

The management, observation, and regulation of the combustor system are centralized through a NI cRIO-9067, an embedded industrial controller specifically designed for such complex tasks. System operations are controlled using a LabView program complemented by a graphical user interface (GUI) shown in [Fig. 4.6](#), ensuring seamless interaction with the data acquisition setup. Various National Instruments (NI) modules are deployed to cater to the unique data acquisition and control needs of different components within the system. Temperature readings from thermocouples are gathered using a NI9213 module, which incorporates proprietary technology for cold-junction temperature compensation. The NI9870 module is responsible for collecting data from the mass flow controllers. Pressure levels of the fuel, air, and diluent sources are tracked through a NI9205 module, and the heating element is managed via a NI9265 module. Additionally, specific modules are dedicated to operating solenoid valves and managing input/output interfaces, which include system warning indicators and other monitoring functions. The assembly of the NI controller and its modules is contained within a control cabinet, as illustrated in [Fig. 4.7](#).

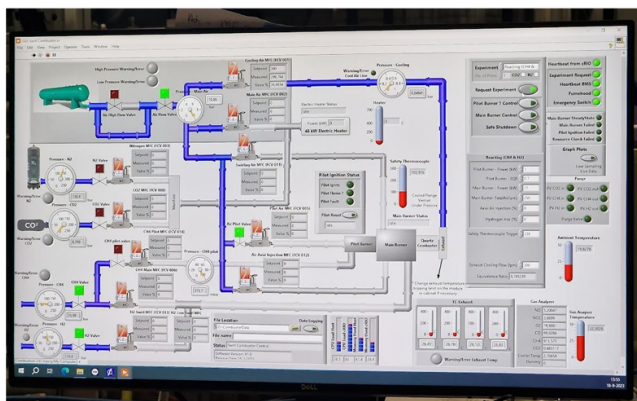


Figure 4.6: Labview GUI used to control the setup



Figure 4.7: Data acquisition and control cabinet holding the cRIO-9067 embedded controller and different NI modules

4.2.6. VENTILATION SYSTEM

The combustor setup is build underneath a fume hood which removes any flammable gases or fumes that escape from the setup. The fume hood remains operational throughout the experiment and continues to run afterwards to comply with ATEX (Atmospheres Explosibles) safety standards. Additionally, the ventilation system, which connects to the gas rack, extends outside the building. The air flow rate through the fume hood is approximately 4600 cubic meters per hour, while the gas rack's ventilation manages around 1650 cubic meters per hour.

4.3. FORCED EQUIVALENCE RATIO FLUCTUATION SYSTEM

This section discusses the subsystem added to the combustor to impose necessary equivalence fluctuations on the flame for the purpose of this research. In order to allow for the advanced modulation of the flow part of the inflows needed to be redirected and altered by a high frequency valve as presented in [Subsec. 4.3.1](#). The specifications of the valve are discussed in [Subsec. 4.3.1.1](#), whereas the control system is discussed in [Subsec. 4.3.2](#). However, the microphone and pressure sensor readings that are performed through the same system are discussed in the methodology chapter, more specifically [Sec. 5.2](#).

4.3.1. OVERVIEW VALVE SUBSYSTEM

To effectively simulate and control equivalence fluctuations in air flow, a decision was made to develop a separate system for safety and practicality. This dedicated system is detailed in [Fig. 4.8](#) and initiates from the combustor lab's shop air supply. This approach was taken to secure an independent air supply and to regulate the main air flow without diverting a portion of it. The shop air system supplies dry compressed air at a pressure of 10 bars. A pressure regulator is installed at the system's output to ensure the air is at the same atmospheric pressure under which the combustor operates and the components are calibrated.

Air then passes through a rotameter, which has been adjusted for atmospheric conditions, allowing the flow rate to be controlled between 0 to 30 liters per minute (lpm). This adjustment is crucial for aligning the supplemental flow with the main flow, ensuring the total air flow remains consistent. Subsequently, a high-frequency valve introduces regular flow fluctuations by alternating between open and closed states at a user defined frequency.

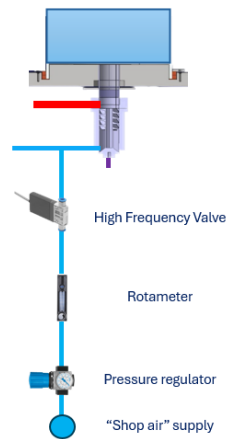


Figure 4.8: Overview of modulation subsystem

VALVE

The key component in the subsystem is the high frequency valve that modulates a small portion of the flow and imposes fluctuations on it. A brief market study was conducted to find the most optimal solution; this eventually led to the choice for the Festo MHJ10 high frequency valve series that can reach speeds well above the required 200 Hz that is needed for the experiments. This frequency requirement stems from the lowest natural frequencies occurring in the system, i.e. thermoacoustic (160 Hz) and non-reacting flow PVC (200 Hz). A close-up of the valve is provided in Fig. 5.4 and the most important specifications are given in Table 4.3.

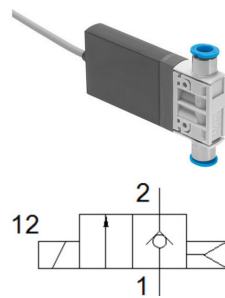


Figure 4.9: Festo MHJ10 valve

Table 4.3: Valve Specifications for flow modulation

Feature	Value
Function	2/2 closed, monostable
Actuation	Electrical
Max Flow Rate	160 l/min
Pressure Range	0.5 ... 6 bar
Switching frequency	≤ 500 Hz
Switching Time	1.2 ms (open), 0.6 ms (close)
Voltage Range DC	21.6 ... 26.4 V
Trigger Range DC	3 ... 30 V

HIGH FREQUENCY VALVE CHARACTERISATION

In order to characterise the valve flows a simplified numerical counterpart was made in Openmodelica as seen in Fig. 4.10. The valve specifications were modelled allowing one to get a clear view of how the flow inertia at the valve affects through put during operations.

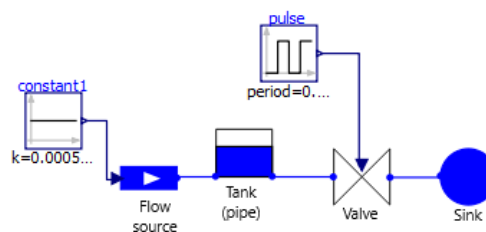


Figure 4.10: Openmodelica model of valve

An important fact that needs to be made clear is that the valve will limit the flow during operations and that this is not compensated by the main air system and hence the flame will become richer on average. This can be attributed to the valve being closed for 50% of the time. With the valve flow rate being set to 25 nlpm this

will have an impact on the setpoints. Additionally, due to the inertia of the flow there will be a build-up of flow and pressure at the valve, however due to the additional rotameter this is quite limited as can be seen in Table 4.4. The effect on the equivalence ratio depends on the enrichment level and is discussed in ??.

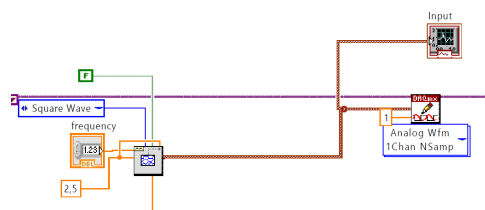
Table 4.4: Valve output flow characteristics after 10 seconds of modulation

Valve output flow characteristics		
Standard open valve flow characteristics:		
Mass flow:	25 nlpm	Pressure: 1.013 bar
Modulated flow characteristics*:		
f	$\Delta \dot{m}$	Δp
All	0.1%	0.0015%

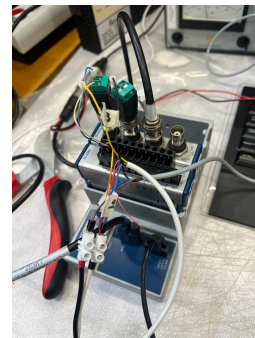
*After 10 seconds of modulation

4.3.2. VALVE SUBSYSTEM CONTROL AND ACQUISITION

The control centre of the subsystem is centred around a National Instruments (NI) Compact Data Acquisition system (NI cDAQ-9174), which is equipped with both an analogue input (NI-9215) and an analogue output (NI-9263) module, as illustrated in Fig. 4.11b. The analog input module is tasked with capturing signals from acoustics, pressure, and the raw valve input signal, while the analog output module facilitates the actuation of the valve through a 10-volt signal. The operational logic for data acquisition and valve actuation is governed by LabView programming, crafted specifically for these experiments. A snippet of this code is showcased in Fig. 4.11a, with the complete code available in Appendix D. This programming enables the generation of a square wave signal, which is adaptable via pulse-length modulation (PLM)—allowing for adjustments in both the duty cycle and frequency. As a result, the valve is programmed to open at the peak of the wave and close at its trough or upon loss of the power signal. Throughout the experimental setup, the duty cycle is maintained at 50%. The LabView control board built for the experiments, depicted in Fig. 4.12, provides a direct visual of the measurements and facilitates straightforward valve manipulation.



(a) Snippet of LabView code



(b) CDAQ controlling the system

Figure 4.11: Valve subsystem control

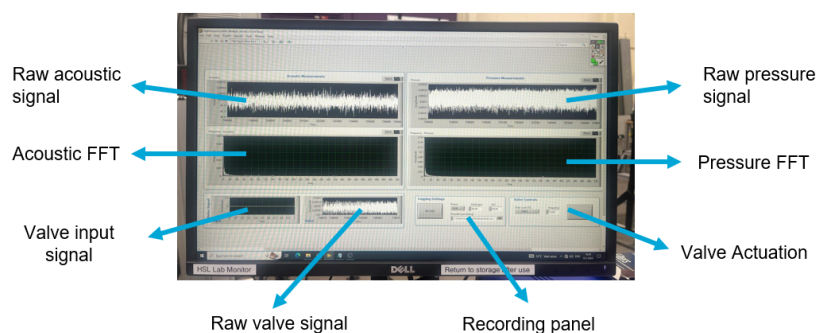


Figure 4.12: Valve control panel

5

METHODOLOGY

In this chapter, the methods employed to capture the data are detailed. The data used to address the research questions were obtained across a variety of set points covering a wide range of operating conditions within the swirl combustor. These set points featured flames varying in power level (7 to 12 kW), volumetric hydrogen enrichment (0 – 100%), and axial air injection (10 – 20%). Each set point was analyzed under its baseline, non-actuated condition and at least two or more different actuated conditions. The methodology for defining and obtaining these set points is discussed in [Sec. 5.1](#).

Each setpoint was captured using a multitude of different measurement techniques that were already introduced in the literature study and focus on different aspects of the flame. The exact set-up and specifications for each measurement are discussed in the subsequent sections. Starting with the acoustic and pressure measurements in [Sec. 5.2](#). Followed by the emission measurements in [Sec. 5.3](#) and chemiluminescence measurements in [Sec. 5.4](#). Lastly, the velocity measurements and associated flow structure analysis technique are presented in [Sec. 5.5](#), together with an overview in [Sec. 5.6](#).

The data was also captured over several experiential campaigns in the sustainable propulsion lab at the TU Delft aerospace faculty. Since the knowledge regarding the combustor and its behaviour also grew during these test campaigns, the setpoints were also updated and modified during the thesis research. Hence, the test campaign and their impact on the test points are also discussed in [Sec. 5.1](#).

5.1. TEST CAMPAIGNS AND SET-POINTS

As already alluded to in the Introduction, it was decided to spread the measurements over several campaigns. In total, three experimental campaigns were conducted, each with their own purpose and available measuring equipment that met the requirements. The setpoints were chosen in line with the purpose of each experimental campaign that was conducted in collaboration and with the help of S. Link. These will be discussed together with the test matrices devised for each campaign. In each campaign the set-up had to be reassembled and, if applicable, the new additions to the systems had to be tested, characterised and commissioned.

5.1.1. INTRODUCTORY TEST CAMPAIGN SEPTEMBER

The introductory test campaign was led by S. Link and mainly served as introduction to the system and the measuring techniques. The initial campaign focused on the influence of the Reynolds number, and the sensitivity of unstable conditions to small steps in AAI, hydrogen content and equivalence ratio. In this initial shorter campaign only OH^+ -chemiluminescence, farfield acoustic and emission measurements were taken. The testmatrix involved a wide variety in both hydrogen content, AAI percentage and power setting. Moreover, the effect of the swirl number was also varied from 1.1 to 0.7 by swapping out the swirler. It was decided to use a selection of these setpoints as the baseline points for the next campaign. These are shown in [Table 5.1](#). A distinction can be made between low (8 kW) and high (12 kW) power mode. The power setting of the pure hydrogen point was chosen to be inline with the ongoing numerical research into the combustor such the data can be compared. The hydrogen enrichment percentage is achieved by volumetric ratio $\text{hydrogen\%} = (\text{volume}_{\text{H}_2} / \text{volume}_{\text{fuelmix}}) \cdot 100$.

Table 5.1: Test matrix for baseline points

Baseline test matrix				
Group	Power	Air flow	AAI	Volumetric hydrogen content
Low power	8 kW	167 nlpm	10 %	25, 40, 60 %
High power 1	12 kW	250 nlpm	10 %	25, 40, 60 %
High power 2	12 kW	250 nlpm	20 %	0, 25, 40, 60, 80 %
Pure hydrogen	11 kW	200 nlpm	20 %	100 %

Table 5.2: Flame conditions range high power (12 kW)

CH4	H2	equivalence ratio [-]	fuel momentum J [-]	rho mix [kg/m3]	m air [nlpm]	flame speed [cm/s]	flame temperature [K]
1	0	0.75	1.373	0.681	250	24.68	1925
0.8	0.2	0.74	1.418	0.562	250	27.800	1920
0.6	0.4	0.722	1.589	0.4427	250	32.160	1907
0.4	0.6	0.702	1.782	0.3236	250	40.296	1898
0.2	0.8	0.6698	1.9384	0.20449	250	44.790	1885
0	1	0.61	1.682	0.0853	250	93.770	1861

Table 5.3: Flame conditions range low power (8 kW)

CH4	H2	equivalence ratio [-]	fuel momentum J [-]	rho mix [kg/m3]	m air [nlpm]	flame speed [cm/s]	flame temperature [K]
1	0	0.768	1.63	0.667	167	29.12	1977
0.8	0.2	0.759	1.77	0.550	167	33.31	1972
0.6	0.4	0.747	1.93	0.443	167	41.23	1960
0.4	0.6	0.728	2.07	0.341	167	47.77	1935
0.2	0.8	0.698	2.11	0.200	167	53.44	1891
0	1	0.64	1.66	0.0835	167	91.85	1874

5.1.2. FIRST PIV TEST CAMPAIGN OCTOBER

The second campaign focused on gaining insights on the flow dynamics in the reacting flows inside the swirl combustor. More specifically, in the campaign the baseline unmodulated flow measurements were made using high speed PIV set-up detailed in [Sec. 5.5](#). Additionally, the PIV flow seeder was commissioned and characterised within this campaign by J. Lebreun and S. Link. The test matrix was built on the data points selected in the previous test campaign, presented in [Table 5.1](#). The set points were still conducted for both the 1.1 and 0.7 swirl burner.

5.1.3. SECONDARY PIV TEST CAMPAIGN MARCH

The final campaign saw the introduction of the valve subsystem that allows one to modulate part of the flow at a set frequency. This system had to be build, tested and characterised before and at the start of the campaign. The campaign focused on collecting both flow dynamic and emission data of the induced forced equivalence ratio fluctuation of the standard baseline points established in the previous campaigns and gaining insight into the valve subsystem. Therefore, only the modulation magnitude and frequencies had to be determined for the test matrix.

The magnitude of the modulation, i.e. the relative amount of the total air flow that is modulated by the valve, was determined to be 25nlpm or 10% of the total flow for power 12. Such that the steady state conditions are still fulfilled for those points¹. Additionally, the magnitude was also limited to guarantee safe operations when modulating the flow. Namely, when the valve is closed the equivalence ratio increases and causing the flame speed to increase and increasing the likelihood of flashback. This was checked by calculating the

¹For power 8 kW some additional measurements with 16nlpm or 10%

increase in flame speed for the points in Cantera², which can be seen in Fig. 5.1. The increased flame speed for a 25 lpm decrease in airflow was deemed acceptable. Originally, it was intended to go leaner such that the flame speed becomes slower and the system is in turn safer as can be seen in the stability graph. However, during the experiments the valve was set to decrease the air flow by 10% making the mixture richer. This was realised to late in during the experiments to be corrected and the operations were still deemed safe and stable enough to use these results.

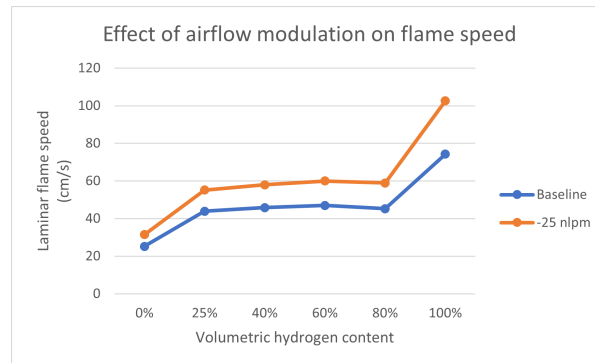


Figure 5.1: Effect for a 10 % decrease in airflow due to valve position on flame speed

Furthermore, the conditions were also checked in the in the stability limits for the set-up established by S. Link. The baseline reference points and the difference in equivalence ratio for an added or a decreased air-flow are shown for both 10% and 20% AAI cases in figure Fig. 5.2. From this graph it is clear that the intended originally decrease in equivalence ratio by increasing the air flow with the valve would have let to safer operating conditions. However, as one can see the actual valve configuration that was used let to a decrease in airflow, and therefore increase in equivalence ratio. This did not turn out to be as big of a problem as the graph might suggest. The effect on the equivalence ratio is presented in Table 5.4 for the low power setpoints and Table 5.5 for the high power setpoints.

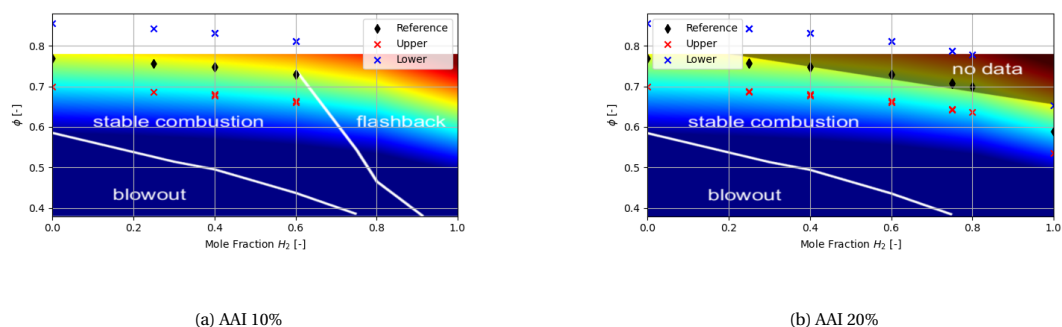


Figure 5.2: Modulated equivalence ratio for a 10 % increase or decrease in airflow due to valve position visualised in stability limits for the set-up established by S. Link

²<https://cantera.org/>

Table 5.4: Flame condition when valve is closed for low power set points (8 kW)

CH4	H2	equivalence ratio [-]	fuel momentum J [-]	rho mix [kg/m3]	m air [nlpm]	flame speed [cm/s]	flame temperature [K]
0,8	0,2	0.89	2.39	0.550	142	49.11	2157
0,6	0,4	0.878	2.57	0,443	142	58.94	2150
0,4	0,6	0.856	2.75	0.341	142	67.41	2126
0,2	0,8	0.821	2.77	0.200	142	74.28	2084

Table 5.5: Flame condition when valve is closed for low power set points (12 kW)

CH4	H2	equivalence ratio [-]	fuel momentum J [-]	rho mix [kg/m3]	m air [nlpm]	flame speed [cm/s]	flame temperature [K]
0,8	0,2	0.84	2.16	0.550	225	42.7	2110
0,6	0,4	0.83	2.34	0,443	225	52.8	2090
0,4	0,6	0.81	2.50	0.341	225	60.6	2056
0,2	0,8	0.77	2.52	0.200	225	66.84	1989

Regarding the determination for the frequency of the valve actuation there was more freedom. Naturally, there were the technical limitations of the valve which had to operate for extended periods for steady state emission measurements further limiting the maximum obtainable frequency of the valve due to overheating concerns. Given that it was the first time this actuation system was used in the lab the commissioning and testing parts were also used as an exploratory phase and helping to establish the frequencies. It was then decided to actuate at the first natural frequency of the chamber, which was known to be 160 Hz. It was also decided to actuate at a lower frequency that is not a multiple of this main mode so 60 Hz and the double of that frequency so 120 Hz. Lastly, it was also decide to actuate the system at 200 Hz, which is the frequency of the PVC that is seen in the cold flow at low power mode (8 kW) by [Link *et al.* \[145\]](#).

However, an unnoticed and unaccounted shift in the valve actuation caused these actuation frequencies to shift to 70 Hz, 140 Hz, 190 Hz and 230 Hz respectively. This was verified with an oscilloscope after the final experiments and the first measurements from the SPOD. Therefore, the complete test matrix for this campaign, given in [Table 5.6](#), has different frequencies from the intended ones. Yet, the measurements are still valid for these actual frequencies.

Table 5.6: Test matrix for modulated points

Modulated test matrix				
Group	Hydrogen content			
Low power	25, 40, 60 %	25, 40, 60 %	25, 40, 60 %	40%
High power 1	25, 40, 60 %	25, 40, 60 %	25, 40, 60 %	40%
High power 2	0, 25, 40, 60, 80 %	0, 25, 40, 60, 80 %	<i>n.a.</i>	<i>n.a.</i>
Pure hydrogen	100 %	100 %	<i>n.a.</i>	<i>n.a.</i>
Valve frequency	70 Hz	140 Hz	190 Hz	230 Hz

5.2. DATA ACQUISITION SYSTEM (DAQ) MEASUREMENTS: ACOUSTICS, PRESSURE, VALVE

The DAQ system, which is utilised for valve actuation, as discussed in [Subsec. 4.3.2](#), is also used to simultaneously capture signals from a directional microphone, pressure sensor, and valve. This is facilitated by an NI-9215 analogue input card, which reads the analogue voltage signals from these various components and digitizes them at a sampling rate of 2000 Hz. Subsequently, these individual signals undergo post-processing and visualisation, typically through a power spectral density spectrum, as exemplified in [Fig. 5.3](#).

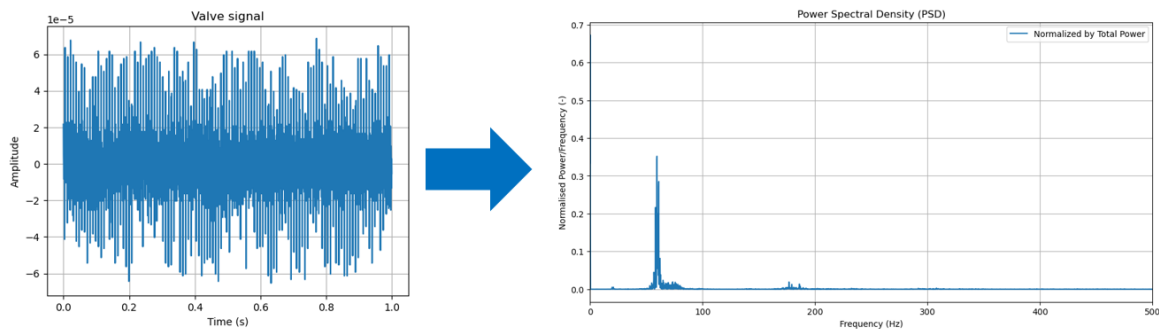


Figure 5.3: Power spectral density from raw signal

VALVE

By attaching an additional cable to both the power and input connections of the valve to the DAQ system, it becomes feasible to directly monitor and verify the voltage input signal sent to the valve. This capability enables correlation between pressure and acoustic readings with the exact valve signal.

PRESSURE SENSOR

The pressure inside the chamber was measured using an industrial pressure transducer (see [Table 5.7](#)) that was connected to the chamber through a short cannula tube through the bottom plate of the burner. The sensor location is shifted 3.5 cm relative to the center of the swirler, as result it mainly measures fluctuations in the outer recirculation zone.

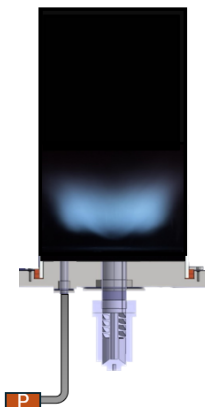


Figure 5.4: Chamber pressure sensor

Table 5.7: Specifications chamber pressure transducer

Chamber pressure transducer specifications	
Model	OMEGA PMX 319 - 0.35G10V
Measuring range	0 - 350 mB
Total error band	$\pm 1.5\%$ gage and absolute
Response time	1 ms
Bandwidth	DC to 1 kHz type
Temperature range	-40 C° - 85 C°
Output signal	15 to 30 Vdc at 10mA

MICROPHONE

A LinearX M53 directional microphone placed approximately 1.5 m from the combustor and directed toward it (as depicted in [Fig. 5.5](#)) is utilised to measure the acoustics of the far field. Refer to [Table 5.8](#) for microphone specifications.

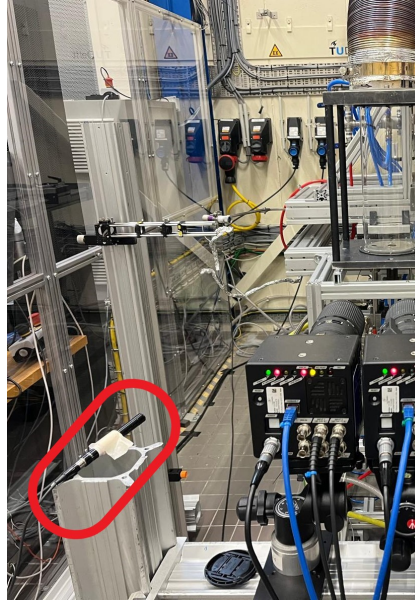


Figure 5.5: Directional microphone pointed at the combustor

Table 5.8: Specifications far field microphone

Far field microphone specifications	
Model	LinearX M53
Element type	Electret condensor
Uncorrected freq response	$\pm 5\text{dB}$ 20 Hz - 20kHz
Acoustic sensitivity	-15dBm/P
Temperature range	0 C° - 40 C°
DC Supply voltage	9 - 11 Vdc

5.2.1. CALIBRATION PROCEDURE DAQ MEASUREMENTS

The measurement tools are calibrated and rated by the manufacturer. First, the mean was subtracted from the signal to obtain the oscillations, and a Hamming window was applied to limit spectral leakage.

5.2.2. MEASUREMENT PROCEDURE

Measurements were made once the conditions of a set point were reached. The three measurements were taken simultaneously for at least 10 s and stored together in a csv file. Background noise from the room was spectrally subtracted from the readings. The Welch method, explained in [Subsec. 2.3.3.1](#), was used to convert the signal to a power spectral density function.

5.3. EMISSION MEASUREMENTS

In order to get an indication of the emissions of a certain set point a sample of the exhaust gases is analysed. The sample is collected through a probe mounted in the exhaust just above the quartz chamber. A heated sample line then transports the sample to the gas analyser system that is shown in [Fig. 5.6](#). This system is a customised ABB AO2000 extractive gas analyser system containing a *Limas 21* , *Uras 26* and *Magnos 26* analyser [95]. A short description of each one is provided below together with an overview of the achieved measurement ranges in [Table 5.9](#), and the working principles were already discussed in [Subsec. 2.3.1.3](#). For a more in depth explanation of the system please refer to the thesis by [Goselink \[177\]](#) who selected the system.

Uras (NDIR):

- Utilizes Non-Dispersive Infrared (NDIR) spectroscopy.
- Specialized for NO_x and SO₂ measurement in exhaust gases.

Limas (NDUV):

- Relies on Non-Dispersive Ultraviolet (NDUV) spectroscopy.
- Used for detecting gases like CO, CH₄, CO₂.

Magnos (Paramagnetic):

- Based on paramagnetic analysis.
- Primarily measures percentage-level oxygen.

Table 5.9: Measuring ranges and accuracies for the different emission species

Species	Range	Accuracy
NO	0 - 10 ppm	0.1 ppm
NO ₂	0 - 10 ppm	0.1 ppm
CO	0 - 75 Vol%	0.1 Vol %
CO ₂	0 - 1000 ppm	1 - 10 ppm
CH ₄	0 - 50.000 ppm	0.01 Vol %
O ₂	0 - 25 Vol %	0.1 Vol %



Figure 5.6: Gas analyser with calibration bottles

5.3.1. CALIBRATION PROCEDURE

At the start of each test day the gas analyser must be calibrated using either a span or 2-point calibration as depicted in Fig. 5.7. This involves matching the sensor's output to known values at two points: the 'zero' point (no target gas present) and the span (specified amount of target gas). If there's a discrepancy between the measured and expected values, the system is re-calibrated accordingly.

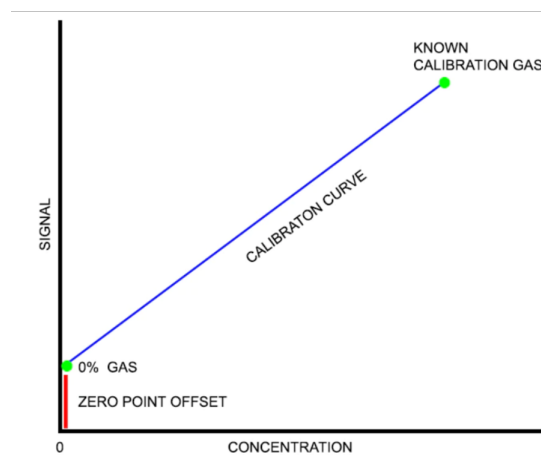


Figure 5.7: 2-Point calibration curve [178]

For the calibration process, specialized gas mixtures calibration bottles are available. These bottles contain a predetermined amount of the target gas, which represents 80% of the measurement span, and nitrogen. These calibration gases have a high level of purity to avoid any components that might cause cross-sensitivity, ensuring the integrity of the calibration. The composition of these gases and their quantity are detailed in Table 5.10. Each batch is tested and certified by the supplier to confirm that the mixture concentrations fall within the acceptable tolerance range of either ± 2 ppm or $\pm 5\%$, depending on the specific gas mixture.

Table 5.10: Calibration bottles and their respective calibrated values

Calibration Bottle	NO	NO ₂	CO	CO ₂	CH ₄	O ₂
NO ₂ + N ₂	-	8.1 ppm	Zero	Zero	Zero	-
CO ₂ + N ₂	Zero	Zero	-	59.8 vol%	-	-
NO + N ₂	8.46 ppm	Zero	-	-	-	Zero
CO + CH ₄ + N ₂	-	-	801 ppm	-	3.99 vol%	-
Ambient Air	-	-	-	-	-	20.95 vol%

Zero calibration is crucial for establishing a baseline reading where no target gas is present. For this, one of the calibration bottles that does not contain traces of the target gas is used, ensuring the accuracy of the zero point setting. During calibration, the analyser creates a calibration curve that assumes a linear relationship between the zero and 80% span points. This curve is critical for the analyser to accurately determine gas concentrations in future measurements, adjusting for any initial discrepancies known as zero-point offsets. This ensures that all subsequent sensor readings are corrected based on the established zero-point data

5.3.2. MEASUREMENT PROCEDURE

The measurements of the gas analyser are sampled at a 10 Hz rate, and sent to the central control computer where they can be monitored live and stored. The data is stored in a TDMS file containing all the monitoring data of the entire combustor system, such as power setting, flow rates, pressures and temperatures.

In the initial experimental campaign the emission measurements were stored continuously and the data points for the set points had to be extracted manually from the the large data set. This was done with a matlab code that was available and provided in [Appendix E](#). The code has to find the set points within the dataset and determine if steady state is reached. Only then the steady state emission values are calculated.

For the last campaign that looked into the flow modulation a separate measurement was made for each specific set point. For each point the modulation, i.e. the valve, was turned on at the predetermined frequency and left running for 2 minutes such that the system reaches steady state. Only then, a 30 second measurements was made and stored in a dedicated folder. This simplified the post processing significantly, leaving only the oxygen correction to be applied once the emission data was extracted from the file.

OXYGEN CORRECTION

As the amount of excess air varies between the different set-points the relative concentration of the emission will also be affected. In order to correctly compare measurements one needs to place them within the same reference frame. Therefore, it is standard practice in literature to correct the emission measurements for NO_x to a 15% O₂ basis [179]. This can be done using [Eq. 5.1](#) with X_{meas} the measured concentration in vol%, X_{corr} the corrected concentration, O_{2ref} the reference concentration O₂ (in this case 15vol%), and O_{2meas} the measured O₂ concentration in vol%.

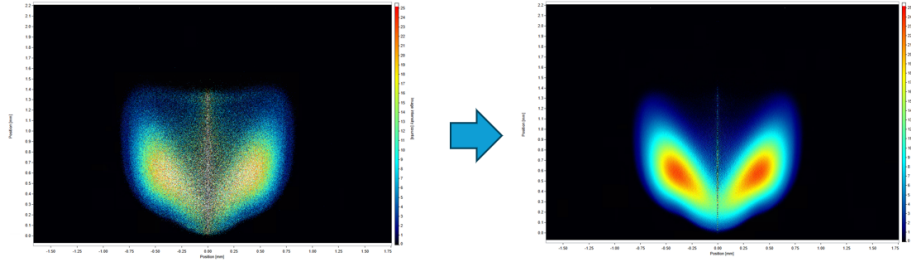
$$X_{corr} = X_{meas} \cdot \frac{(20.95 - O_{2ref})}{(20.95 - O_{2meas})} \quad (5.1)$$

5.4. CHEMILUMINESCENCE MEASUREMENTS

A Tucsen FL-20BW equipped with a 78 mm lens and a narrow 50 nm band-pass filter centered around 325 nm was used to capture OH*-chemiluminescence images. These images assist in identifying and analysing the flame front. The exposure time was adjusted on a case-by-case basis, decreasing as the flame brightness increased. The images were manually calibrated in DaVis using a calibration plate. The camera was focused at the middle of combustion chamber at start of the campaign.

5.4.1. POST-PROCESSING: ABEL DECONVOLUTION

An inverse Abel transformation is required to construct a true image of the flame front from the two-dimensional picture. The Abel inversion or deconvolution allows one to reconstruct the intersection of the three-dimensional swirling flame from the two-dimensional image. [?] After the transformation a 5x5 smoothing function was applied to the images. Then an average frame was constructed from the subset of images for each measuring point.

Figure 5.8: Abel transformation of OH^* chemiluminescence

5.4.2. MEASUREMENT PROCEDURE

For each set point, the digital exposure was adjusted while the gain remained unchanged. Subsequently, a series of 10 images was captured. These were then processed later in DaVis as described in the previous section.

5.5. VELOCITY MEASUREMENTS

In this section, the PIV setup and measurement procedures are described. First the setup is explained in [Subsec. 5.5.1](#). Then the calibration and measurement procedures are explained in [Subsec. 5.5.2](#) and [??](#), respectively. Lastly, post-processing and data collection are discussed in [Subsec. 5.5.4](#).

5.5.1. PIV SET-UP

The high-speed PIV setup is seen in action in [Fig. 5.9](#). It consists out of a high powered laser that is used to construct a planar laser sheet through the combustor detailed in [Fig. 5.10](#). This 2 mm sheet is constructed using various optics and runs through the center of the chamber as can be seen in the image. Next, one needs to add the TiO_2 particles in the flow using a seeder. Perpendicular to the sheet, a high speed camera with a green light filter is installed to capture the movement of the particles in the flow from which the instantaneous velocities are eventually calculated. A high-speed PTU (Programmable Timing Unit), more specifically Lavisions PTU x, and lab computer are needed to streamline the timing between the laser and the camera and capture the images. This was done using LaVision's DaVis 11 software.

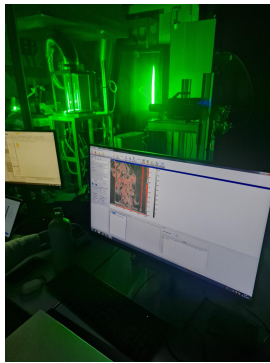


Figure 5.9: PIV setup in action

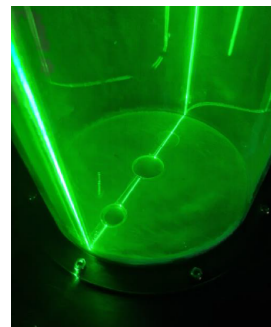


Figure 5.10: Location of laser sheet (in low power mode)

This general PIV setup stayed the same across both test campaigns. However, due to availability both the laser and camera's were changed over the campaigns. The specifications of the laser and cameras for the respective campaigns can be found in the next two subsections.

PIV SET-UP SECOND CAMPAIGN

The PIV setup that was used in the second experimental campaign, i.e. the first PIV campaign, is detailed below. Images showing both the laser and the high-speed camera used are presented in [Fig. 5.11](#) and [5.12](#). The settings of both devices are detailed in [Table 5.11](#) and [Table 5.12](#), respectively. The measurement specifications are presented in [Table 5.13](#).

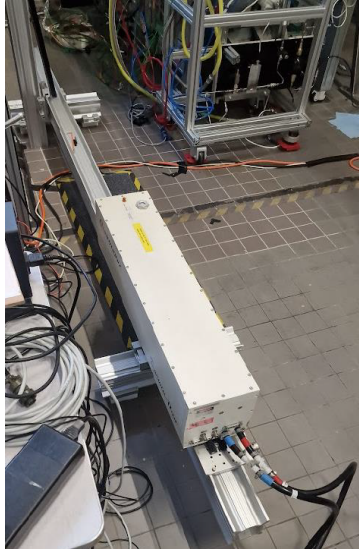


Figure 5.11: Mounted Quantronix Darwin laser



Figure 5.12: LaVision imager HS Pro pointed at setup

Table 5.11: Specifications PIV laser - Second campaign

PIV laser specifications	
Model	Quantronix Darwin Duo 527-80-M
Laser Cavity Material	Nd:YAG
Power	80 W
Wavelength	527 nm
Repetition Rate	0.1 to 10 kHz
Pulse Energy	25 mJ

Table 5.12: Specifications PIV cameras - second campaign

PIV cameras specifications second campaign	
Model	LaVision Imager HS Pro
Maximum fps	1279 fps ^a
Sensor size	2048 x 2048 pixels
Shutter	1ms to 115 ns
Lens	NIKON 200mm AF-D MICRO NIKKOR LENS
Fstop	11
Filter	BP525-52 Light Green Bandpass (500 - 550 nm)

^a Frame rate higher at reduced area of interest

Table 5.13: Specifications PIV measurements second campaign

PIV measurement info second campaign	
Double frames	2000
Frequency	870 Hz
dt	37 μ s
Picture size	10x10cm
Laser current	23.3 A
Tracing particle	TiO ₂

PIV SET-UP THIRD CAMPAIGN

Due to availability both the laser and high speed camera's had to be changed for the last and second PIV campaign. The replacement laser is shown in Fig. 5.13 and its specifications can be found in Table 5.14. The single large high speed camera had to be replaced by two smaller side-by-side cameras as shown in Fig. 5.14. A set-up where these camera's were stacked vertically was also tried but the wider field of was favoured over the taller due to the width of the recirculation zones and the relatively low height of the flame. The details of the cameras can be found in Table 5.15. Due to the limitations of the laser frequency the sampling rate had to be increased to 1000 Hz as indicated in Table 5.16.

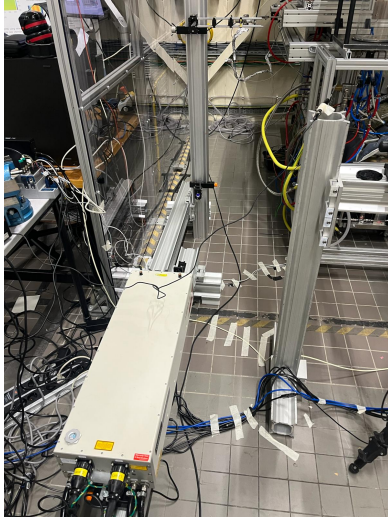


Figure 5.13: Mounted Continuum Mesa laser



Figure 5.14: Side-by-side Photron fastcams capturing the set-up

Table 5.14: Specifications PIV laser - Third campaign

PIV laser specifications	
Model	Continuum Mesa PIV 532-120-M
Laser Cavity	Nd:YAG
Material	
Power	120 W
Wavelength	532 nm
Repetition Rate	1 to 30 kHz
Pulse Energy	18 mJ

Table 5.15: Specifications PIV cameras - third campaign

PIV cameras specifications third campaign	
Model	Photron FASTCAM Mini AX 100
Maximum fps	212,500 fps
Sensor size	1024 x 1024 pixels
Shutter	1ms to 1 μ s
Lens	NIKON 200mm AF-D MICRO NIKKOR LENS
Fstop	16
Filter	BP525-52 Light Green Bandpass (500 - 550 nm)

Table 5.16: Specifications PIV measurements third campaign

PIV measurement info third campaign	
Double frames	2x2000
Frequency	1000 Hz
dt	37-45 μ s
Picture size	2x5.5cm
Laser current	19.9 A
Tracing particle	TiO ₂

5.5.2. CALIBRATION

The images were calibrated in DaVis for the entire campaigns using a set of calibration images taken at the start of the campaign. For the first PIV campaign Fig. 5.15 and the distances were manually inserted. For the second PIV campaign, Fig. 5.16, a calibration plate was available and could be seen by both cameras.

The laser pulse off-set with the camera triggering and the seeding density were found experimentally at the start of the campaign. The seeding density had to be increased with power and hydrogen content during the campaign.

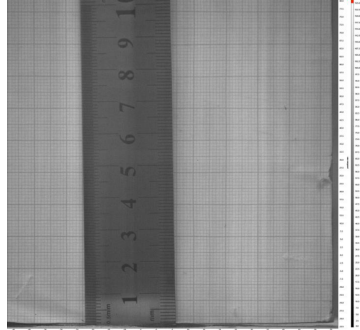


Figure 5.15: PIV Calibration second campaign

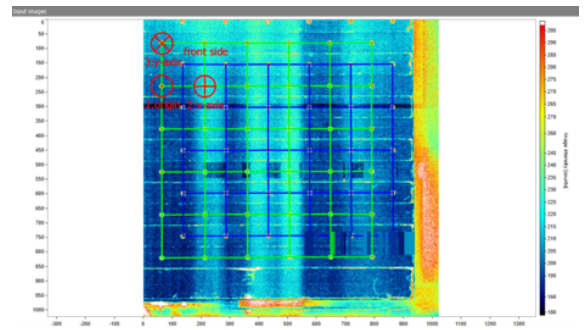


Figure 5.16: PIV Calibration third campaign

5.5.3. MEASUREMENT PROCEDURE

The measurements entails the capturing of 2000 double frames, i.e. two snapshots that are only a few microseconds apart, by the high speed camera synchronous with the laser burst at a set frequency. Due to the different set-ups the specifications varied between the campaigns. For the exact PIV measurement specifications, please refer to Table 5.13 and 5.16 respectively. This is done for the various set points detailed in Subsec. 5.5.1 allowing for the comparison of the effects of hydrogen addition and power.

5.5.4. PIV POST-PROCESSING

The raw images of the illuminated particles were further post-processed in DaVis 11 to obtain the instantaneous velocity profiles. The following steps were undertaken:

1. Pre-processing:
 - Sliding average background subtraction (sliding window: 9 frames)
2. PIV calculation
 - Spatial resolution of 32px
 - Multi-pass:
 - Initial 64x64 px window (50% overlap)
 - 3 x 32x32 px window (50% overlap)
 - Removal of vectors with correlation < 0.5
 - Median filter (5x5 window) for outlier detection and removal
3. Stitching of side-by-side images if applicable
4. Trimming the edge to obtain a square window

AVERAGE FLOW FIELD

A first action one can under take when having obtained the instantaneous velocity profiles is to combine them into an single average velocity images. The reveals the core structure of the flame and removes temporal fluctuations as shown in Fig. 5.17. This can also be done within Davis 11.

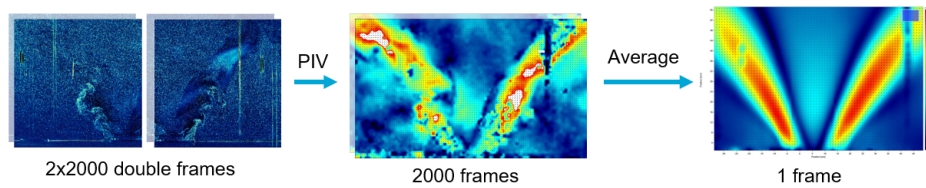


Figure 5.17: Average velocity field procedure

SPECTRAL PROPER ORTHOGONAL DECOMPOSITION

The time-resolved instantaneous velocity fields obtained by the PIV can also be used to uncover some of the underlying coherent flow structures through a Spectral Proper Orthogonal Decomposition (SPOD) [99] of the data. The technique was already explained earlier in ??.

The instantaneous velocity fields were exported from DaVis into Matlab where an open-source SPOD code [180] was adapted to calculate the SPOD. The mean was subtracted from the data and a temporal hamming window dividing the snapshots into segments of 400 frames with 50% overlap was used. This resulted in a temporal resolution of roughly 2 Hz. Then the Fourier transform is applied to the data and the SPOD is calculated.

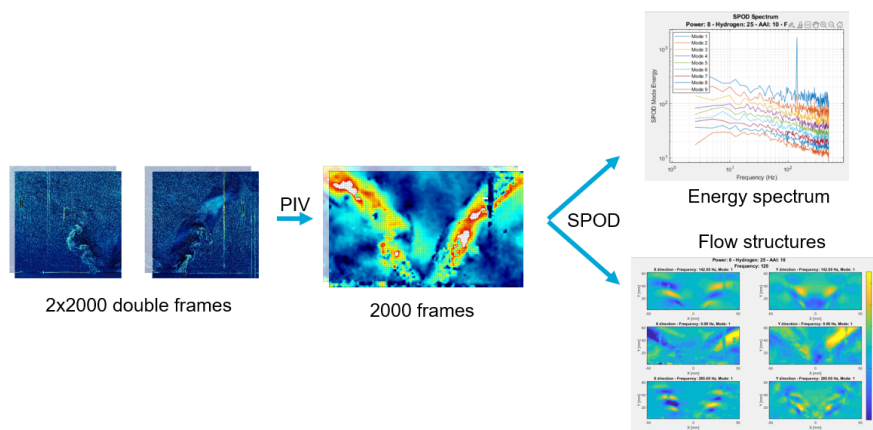


Figure 5.18: SPOD procedure

5.6. SUMMARY OF MEASUREMENTS

An overview of the different measurements made during the campaigns is provided in Table 5.17. The table comprehensively shows what is measured, how it is measured, what the measurement entails, and at what sampling rate the measurement is taken.

Table 5.17: Overview measurement instrumentation specifications

Technique	Specification	Sampling Rate	Measurements	Clarification
Acoustics ^{a,b}	Microphone	2 kHz	10 sec/Meas.	High frequency microphone placed in the far field (1.5 m)
Chamber Pressure	Pressure sensor	2 kHz	10 sec/Meas.	Pressure transducer connected via cannula through bottom plate
Valve actuation	Valve signal	2 kHz	10 sec/Meas.	Voltage of input signal used to trigger valve
Chemiluminescence	OH [*]	15 Hz	10 frames/Meas.	UV camera with 325 ± 50 nm Filter (~1.5 m)
DSLR ^{b,c}	NA	Variable	10 frames/Meas.	ISO 32000 Exposure variable
Exhaust Gas ^a Analyser	CO, CO ₂ , CH ₄ NO, NO ₂ , O ₂	10 Hz	30 sec/Meas.	URAS, LINAS and MAGNOS gas analysers by ABB
PIV1 ^{a,c}	2D2C	870 Hz	2000 sets/Meas. 2 frames/set	2D Velocity measurements using Darwin laser
PIV2 ^{a,c}	2D2C	1 kHz	2000 sets/Meas. 2 frames/set	2D Velocity measurements using Mesa laser

^a Not compatible

^{b,c} Not simultaneous

6

EXPERIMENTAL RESULTS AND ANALYSIS

This chapter will present the different measurements of the various set points. The measurements will be compared and analysed with respect to each other. First, the unmodulated flames will be covered in [Sec. 6.1](#). Then, the effects of the forced equivalence ratio fluctuations will be discussed in [Sec. 6.2](#).

6.1. EFFECTS OF HYDROGEN ENRICHMENT

In this section the results of the reference setpoints, i.e. those with no actuation, are presented. This will be combined with an exploration of the effects of hydrogen enrichment and power setting within the various results. First, the complete measurements of the initial reference point are presented in [Subsec. 6.1.1](#) to give one an overview of the data that has been collected for each point and to describe the correlation between the different measurement types. The effects of enriching this point with hydrogen are then shown and discussed in [Subsec. 6.1.2](#). Lastly, the effect of power is discussed in [Subsec. 6.1.3](#).

6.1.1. DETAIL ON REFERENCE SET POINT

In this section a complete overview of the results for the reference set point are given. The settings for this point can be found in [Table 6.1](#). This point was chosen as it has the lowest power setting and level of hydrogen enrichment. Furthermore, the point contains some key features, like the flow field, flame shape and acoustic structures, that comeback in all the other set points. The flame is stable in the sense that it is not flashing back, however it was chosen because it shows loud thermoacoustic instabilities.

Table 6.1: Key specifications of unactuated low power, low hydrogen content reference point

Specifications baseline reference point			
Power	8 kW	AAI	10 %
Hydrogen	25 %	Air	167 nlpm

THE FLAME

The flame, which can be seen in [Fig. 6.1](#), shows what a typical lean swirling methane flame looks like. The swirling flow structure is clearly apparent within it. This structure also comes back in the OH^* chemiluminiscence images, showing the flamefront, that is presented in [Fig. 6.2](#). It is clear that the flame is situated in the inner shear layer between the swirling jet and inner recirculation zones.



Figure 6.1: DSLR picture of ref. flame
(P:8 kW, H₂: 25%, AAI:10%)

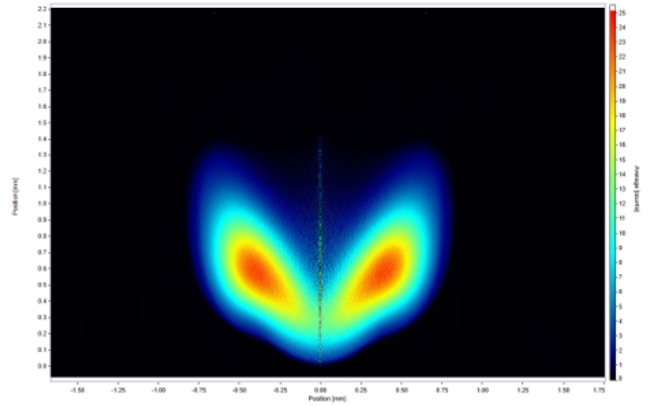


Figure 6.2: OH^* chemi-luminescence of reference flame

AVERAGE FLOW FIELD

The flame location in the shear layer between the recirculation zones is confirmed by the average flow field obtained with the PIV images shown in Fig. 6.3. One clearly sees both recirculation zones separated by the high velocity swirling jets.

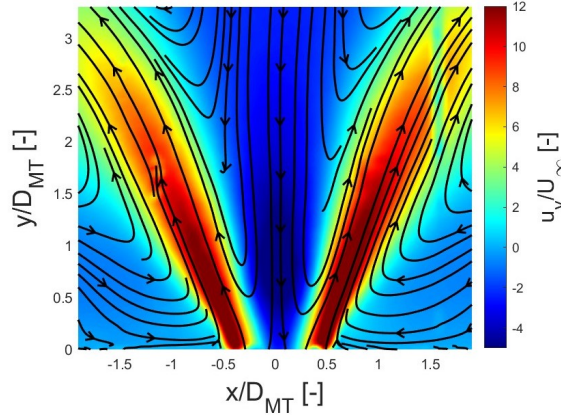


Figure 6.3: Average flow field of reference flame
(P:8 kW, H₂: 25%, AAI:10%)

FLOW FIELD STRUCTURES AND DYNAMICS

In order to gain insight into the underlying flow structures, one can consult the SPOD analyses of the PIV velocity data. For this case, the energy spectrum is shown in Fig. 6.4 and displays a clear peak at 160 Hz. A closer inspection of the flow structure at this frequency in the first mode, presented in Fig. 6.5 reveals an asymmetric pattern in the radial velocity component and a symmetric pattern in the axial velocity component. These patterns are associated with thermoacoustic instabilities [126, 181].

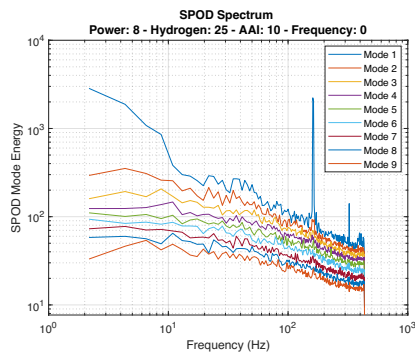


Figure 6.4: Energy spectrum SPOD of ref. flame
(P:8 kW, H₂: 25%, AAI:10%)

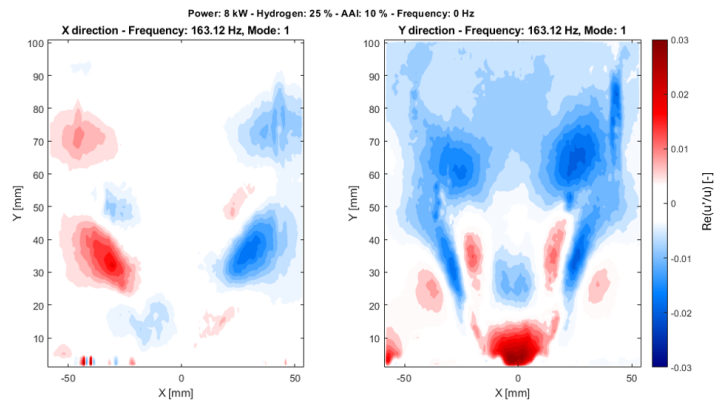


Figure 6.5: Flow structure at peak frequency in SPOD of reference flame
Radial velocity (left) and axial velocity (right) fluctuations at 165 Hz

Precessing Vortex Core

Earlier cold flow tests, conducted by [Link *et al.* \[145\]](#), showed that a strong PVC was prevalent across all conditions. In reacting conditions this PVC is completely suppressed and only some remnants remain of the structure as seen in [Fig. 6.6](#). This phenomenon has been reported by several other researchers in their configurations as well [82]. [Oberleithner *et al.* \[82\]](#) attributes the suppression to the strong radial density gradient at the flame location resulting in a more stable hot wake flow.

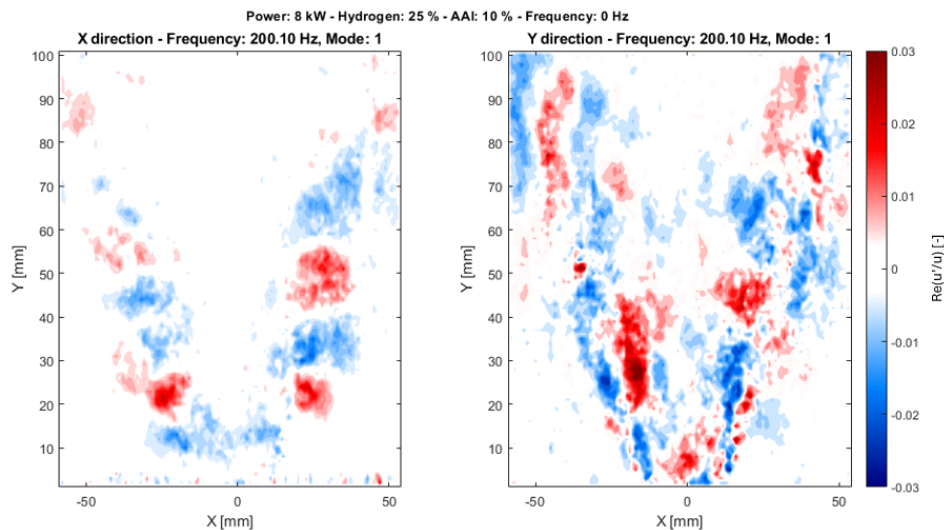


Figure 6.6: Remnants of PVC
Radial velocity (left) and axial velocity (right) fluctuations at 200 Hz
(P:8 kW, H₂: 25%, AAI:10%)

PRESSURE MEASUREMENTS

The chamber pressure and far-field acoustic measurements confirm that the peak in the velocity energy spectrum and associated flow structure originate from thermo-acoustics, as they also possess a strong peak at this frequency in their respective power density spectrum shown in Fig. 6.7.

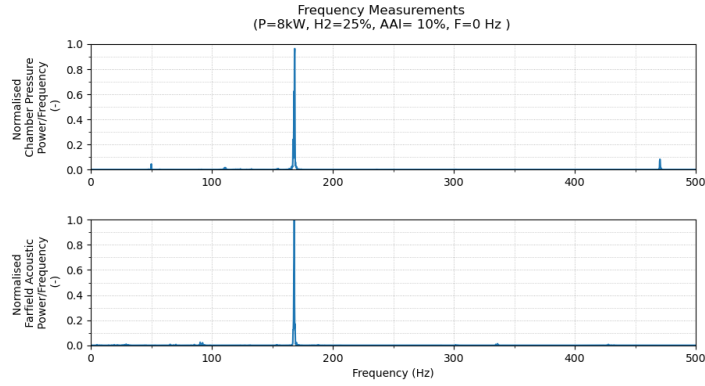


Figure 6.7: PSD of chamber pressure (upper) and far-field acoustics (lower) of reference flame (P:8 kW, H₂: 25%, AAI:10%)

EMISSIONS

Lastly, the final measurements taken for each set point include the concentrations of CO₂ and NO emissions. These specific gases were chosen due to practical considerations: carbon monoxide (CO) is present in such small quantities that reliable measurement is challenging. Nitrogen dioxide (NO₂), on the other hand, poses difficulties because it can condense under certain conditions, complicating accurate measurement. Therefore, focusing on CO₂ and NO allows for more consistent and reliable assessment of emissions across different experimental conditions. For the reference point these came down to 3.5 ppm NO¹ and 3.14 %vol of CO₂².

¹Wet volume ppm at 15% O₂

²wet volume %

6.1.2. EFFECT OF HYDROGEN ENRICHMENT

The addition of an increasingly larger hydrogen fuel fraction has significant effects on both the flame dynamics, stability, and emissions. This is done up to 60% hydrogen enrichment due to the operation limits in low-power mode (8 kW). First, the effects on the average flow field and the location of the flame are analysed in Subsec. 6.1.2.1. This is followed by a look at the effects on the thermoacoustics of the flame together with the temporal flow structures in Subsec. 6.1.2.2. Lastly, the measured effects on the emissions are discussed in Subsec. 6.1.2.3.

AVERAGE MEASUREMENTS

The impact of hydrogen enrichment on the flame location and flame front is made apparent when looking at the OH^* -chemiluminescence figures in Fig. 6.8. It is clear that the flame region becomes smaller and moves closer to the premixing tube, where the mixture enters the combustion chamber. This is inline with the higher expected flame speed and higher reactivity of the enriched mixture.

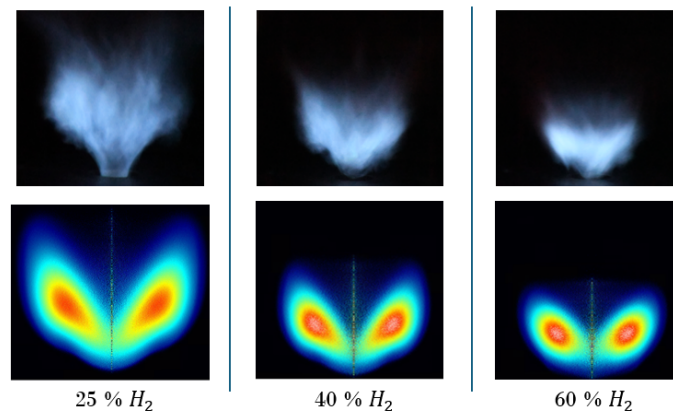


Figure 6.8: Comparison of flame front location and shape for different enrichment ratios
Visual of flame (upper row) and averaged Abel deconvoluted OH^* -chemiluminescence (lower row)
(P:8 kW, H₂: 25 - 60 %, AAI:10%)

Looking at the average axial velocity profiles in Fig. 6.9, one sees that the characteristics elements of the swirling flame, i.e. the swirling jets and the re-circulation zones, are preserved at these hydrogen levels and power setting. It can be noted that the strength of the inner recirculation decreases as the lower flame disrupts the swirling flow earlier and accelerates the flow axially.

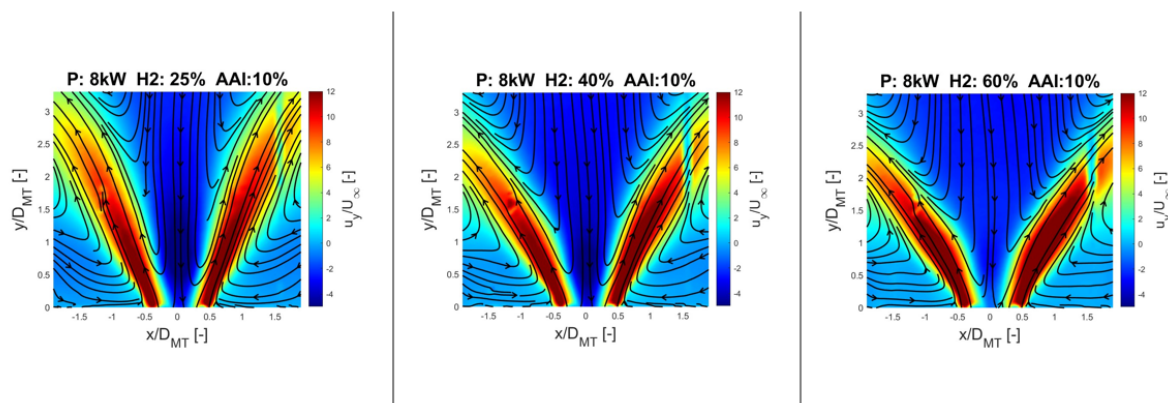


Figure 6.9: Comparison of average velocity fields enrichment ratios
(P:8 kW, H₂: 25% (left), 40% (middle), 60 % (right), AAI:10%)

ACOUSTIC AND TEMPORAL FLOW MEASUREMENTS

Even though the swirling structure prevails for higher hydrogen contents in the low power setting, there is a noticeable effect on the acoustics and in the the flow structures. As illustrated by both the external acoustic

and internal pressure power density spectrum's shown in Fig. 6.10, the thermoacoustic peaks visible in the spectrum are affected by the addition of hydrogen. The initial thermoacoustic, that was apparent at 25% hydrogen enrichment in the low power mode, prevails at 40% enrichment but completely disappears for 60%. The changes can be attributed to the change in composition of the combustion gases flow through the chamber. This different medium density inside the tube will alter its eigen frequency. [182] Additionally, the change in temperature and the general heating of the structure will also influence this. This, in combination with the different flame locations, likely also lies at the basis of the disappearance of the thermoacoustic instability for 60% hydrogen enrichment.

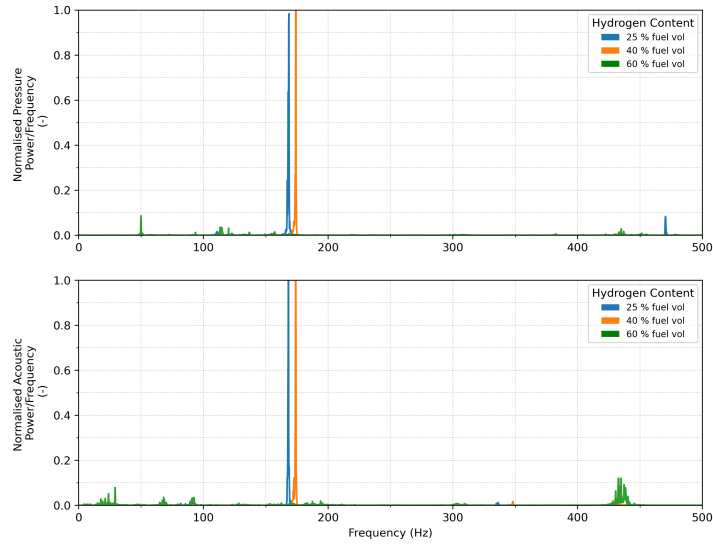


Figure 6.10: Comparison of acoustic measurements for different enrichment ratios
Chamber pressure PSD (left) and far-field acoustics (right)
(P:8 kW, H₂: 25 - 60 %, AAI:10%)

Like for the case enriched with 25% hydrogen, the thermoacoustic peak has a corresponding frequency peak in the energy spectrum of the flow field as revealed by the SPOD in Fig. 6.11. The associated anti-symmetric structure in the flow field of the 40% enriched hydrogen point is similar as that of the baseline case as illustrated in Fig. 6.12.

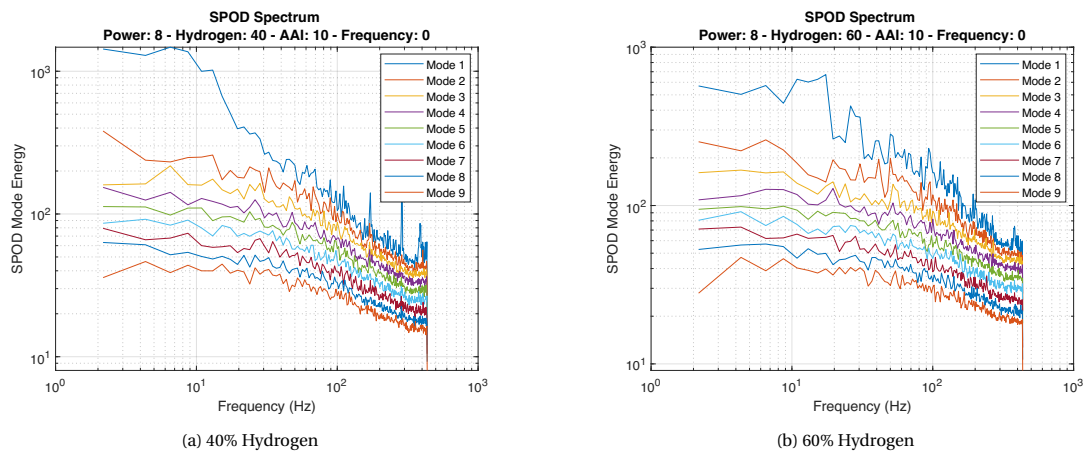


Figure 6.11: Comparison of SPOD energy flow field analysis for different enrichment ratios
(P:8 kW, H₂: 40 - 60 %, AAI:10%)

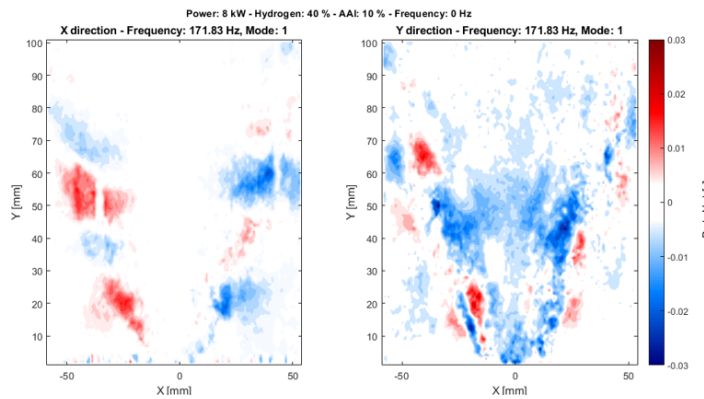


Figure 6.12: Flow structure at peak frequency in SPOD
Radial velocity (left) and axial velocity (right) fluctuations at 165 Hz
(P:8 kW, H₂: 40 %, AAI:10%)

EFFECT ON EMISSIONS

The NO_x emissions, presented in Fig. 6.13, are seen to increase with hydrogen enrichment. Even though, the temperature and equivalence ratio are decreasing with hydrogen enrichment, as per Sec. 5.1, the NO emissions still rise. Looking at the different NO formation pathways introduced in Subsec. 2.1.1.2 it is clear that this rise can be attributed to the NNH route as the thermal and prompt mechanisms decrease due to lowering temperature and carbon levels.

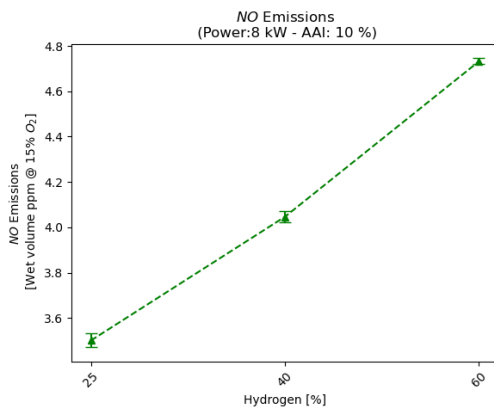


Figure 6.13: Comparison of NO emissions
for different enrichment ratios
(P:8 kW, H₂: 25 - 60 %, AAI:10%)

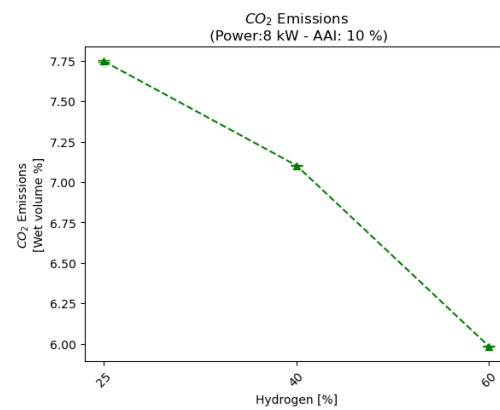


Figure 6.14: Comparison of CO₂ emissions
for different enrichment ratios
(P:8 kW, H₂: 25 - 60 %, AAI:10%)

Due to the decreasing ratio of carbon-based fuel, the CO₂ emissions are expected to decrease with increasing hydrogen enrichment. This is also what was measured throughout the experiments, as seen in Fig. 6.14, and it is the main reason for implementing hydrogen enrichment.

TAKEAWAYS HYDROGEN ENRICHMENT

The analysis of hydrogen enrichment effects on flame dynamics, stability, and emissions reveals several critical impacts. With up to 60% hydrogen enrichment at a low-power setting of 8 kW, significant changes are observed.

Flame Dynamics and Stability:

- **Average Measurements:**
Hydrogen enrichment shifts the flame closer to the premixing tube, reducing the flame region size, aligning with higher flame speeds and reactivity. The swirling jets and recirculation zones are maintained, although the inner recirculation strength decreases due to earlier disruption by the lower flame.
- **Acoustic and Temporal Flow Measurements:**
The addition of hydrogen alters the thermoacoustic peaks and flow structures. At 25% enrichment, thermoacoustic instabilities are present, which continue at 40% but disappear at 60%, likely due to changes

in combustion gas composition, medium density, and flame location affecting the chamber's eigenfrequency and temperature dynamics.

Emissions:

- **NO_x Emissions:**
Contrary to the expectations of decreasing NO emissions due to lower temperatures and equivalence ratios, the NO levels rise with hydrogen enrichment. This increase is attributed to the NNH formation pathway, as the thermal and prompt mechanisms are less active.
- **CO₂ Emissions:**
As expected, CO₂ emissions decrease with higher hydrogen fractions, due to reduced carbon-based fuel content. This reduction is a primary driver for adopting hydrogen enrichment.

In conclusion, hydrogen enrichment significantly impacts flame behavior, acoustics, and emissions. While it enhances flame stability and reduces carbon emissions, it also introduces complex changes in acoustic behavior and increases certain emissions.

6.1.3. EFFECT OF POWER SETTING

At a higher power setting, i.e. 12 kW, one sees similar effects with regards to hydrogen enrichment in both the flow field as well as the acoustic and pressure fields. Namely, the acoustics and emissions are both strongly affected by the increase in power due to the changing flame location and size. The same order as that used for the hydrogen enrichment section is used. Thus the average flow field and flame location are discussed in Subsec. 6.1.3.1, the temporal measurements in Subsec. 6.1.3.2 and the emissions in Subsec. 6.1.3.3.

AVERAGE MEASUREMENTS

A similar story with regards to hydrogen enrichment unfolds at the higher power setting as can be seen in Fig. 6.15 for which a much broader range of operating conditions is possible as the AAI percentage was also increased to 20 %. The OH^{*}-chemiluminescence figures show once again that hydrogen content will cause the flame to become more compact and move toward the premixing tube. The general flames and shapes are similar to the flames seen in Fig. 6.8.

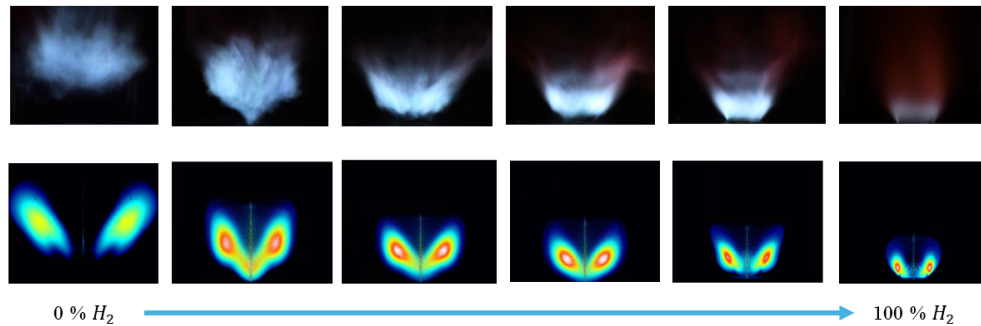


Figure 6.15: Comparison of flame front location and shape for different enrichment ratios.
Visual of flame (upper row) and averaged Abel deconvoluted OH^{*}-chemiluminescence (lower row)
(P:12 kW, H₂: 0 - 100 %, AAI:20%)

The average axial velocity profiles for different enrichment ratios at this higher power setting, shown in Fig. 6.16, tell a similar story as in the low power setting. Once again the characteristic elements of the swirling flame, i.e. the swirling jets and the re-circulation zones, are preserved at these hydrogen levels and power setting. Once again, it can be noted that the inner recirculation zone decreases in strength with increasing hydrogen enrichment. Solely, at pure hydrogen conditions the flow field is significantly different and jet like. This can be attributed to the fact that the flame sits so low and close to the premixing tube that it completely disrupts the swirling flow and turns it into a jet. When you compare the high powered flow fields to the earlier presented low power, in Fig. 6.9, the increase in AAI, which is necessary to prevent flashback, jumps out. The earlier V-shape in the flow field looks more like a W-shape.

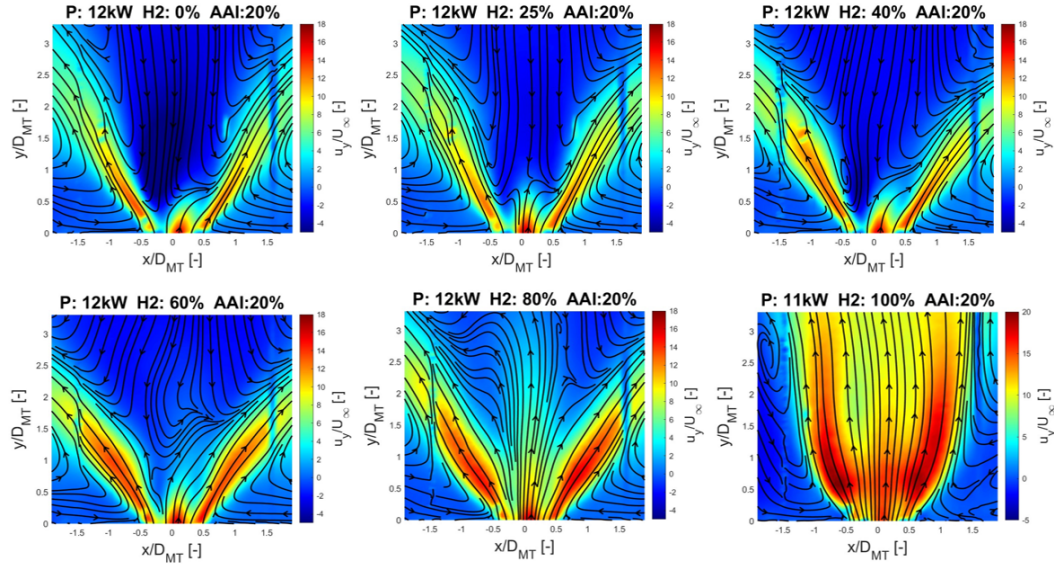


Figure 6.16: Comparison of average velocity fields enrichment ratios
(P:12 kW, H2: 0 - 100 %, AAI:20%)

ACOUSTIC AND TEMPORAL FLOW MEASUREMENTS

Analyzing the acoustic spectrum in the high-power setting, as illustrated in Figure 12, reveals a pattern similar to that observed in the low-power mode. Specifically, the peaks of the spectrum, which correlate with thermoacoustic frequencies, vary with the amount of hydrogen used. For instance, a peak at 450 Hz is observed with 60% hydrogen, whereas peaks at 170 Hz and 350 Hz occur with 40% hydrogen. This indicates that hydrogen enrichment significantly affects both the acoustics and the coherent flow structures within the system.

Moreover, comparing to the peaks in the acoustic spectrum of the low power-setting, presented earlier in Fig. 6.10, which were mainly situated at 160 Hz one sees that they differ from the ones measured in the higher power setting. Hence, it is concluded that the power setting must also have an influence on the thermoacoustics. It's likely that the reasoning for this also lies in the different flame size, flame location, AAI and increased temperature of the air inside the combustion chamber.

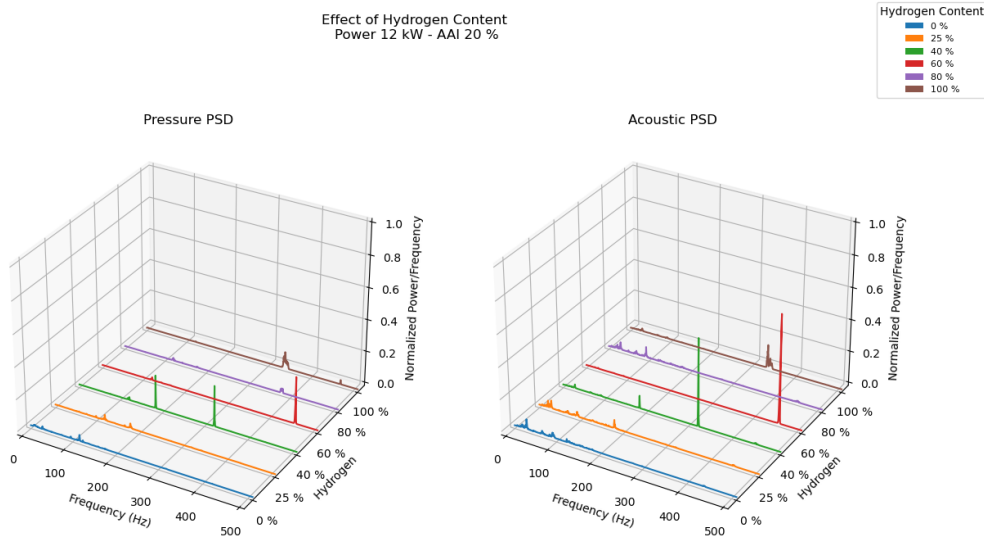


Figure 6.17: Comparison of acoustic measurements for different enrichment ratios
Chamber pressure PSD (left) and far-field acoustics (right)
(P:12 kW, H2: 0 - 100 %, AAI:20%)

That the effect changes in acoustic response for these higher power settings can not be solely explained by a higher AAI percentage is clear when the power of the original low power conditions, presented in Fig. 6.10, is increased without increasing the AAI. These results, presented in Fig. 6.18, still show different acoustic frequencies from the low power mode that also differ from the high power mode with axial air injection.

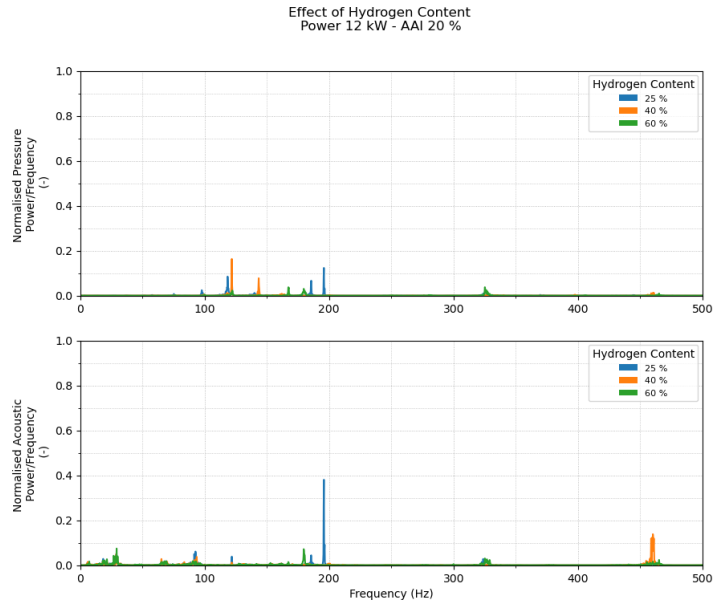


Figure 6.18: Comparison of acoustic measurements for different enrichment ratios
Chamber pressure PSD (upper) and far-field acoustics (lower)
(P:12 kW, H₂: 25 - 60 %, AAI:10%)

EFFECT ON EMISSIONS

Naturally, in a higher power setting, achieved by increasing the respective mass flows, more fuel will be burnt and therefore the absolute CO₂ and NO emissions will also increase. However, the relative measurements, be it *ppm* or *vol%*, should remain the same as the total air volume also increases. A difference in these relative quantities would indicate that the higher-power setting would result in more or less relative emissions. Regarding the emissions CO₂, presented in Fig. 6.20, one sees a slight increase but no difference in trend for the particular emission. This small increase can be attributed to the mixture being slightly richer for the higher power setting, as indicated in Subsec. 5.1.3.

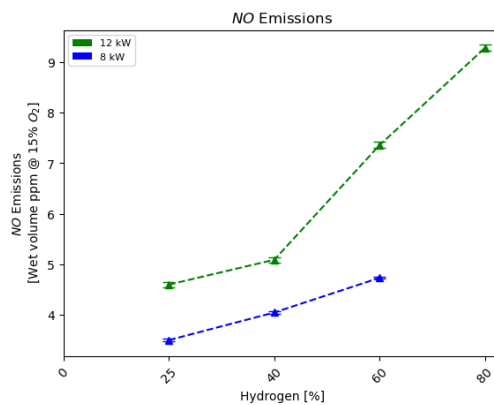


Figure 6.19: Comparison of NO emissions
for different enrichment ratios and power levels
(P:8 - 12 kW, H₂: 25 - 60 %, AAI: 10 - 20%)

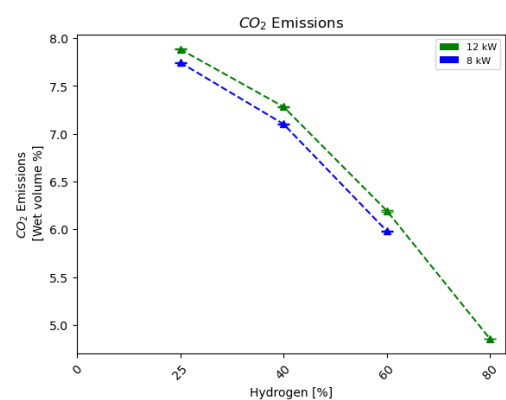


Figure 6.20: Comparison of CO₂ emissions
for different enrichment ratios and power levels
(P:8 - 12 kW, H₂: 25 - 60 %, AAI: 10 - 20%)

The NO emissions paint a different picture. The emissions for the high power setting lie clearly higher than the low power emissions. Moreover, one can see a break in the trend line at 60% enrichment with an even sharper rise in NO emissions then for the low power case. The difference with the lower power setting can

be attributed to the higher peak temperature in the combustion chamber producing more thermal NO. The higher peak temperature is a result of the greater thermal output of the engine, which will increase the temperatures inside the chamber given that it is not actively cooled. Additionally, the increase in fuel will likely result in larger rich pockets within the flame. Hence, the NNH pathway will also produce more NO. Most likely, the latter lies at the base for the trend break as the NO production rises with rich H-radical pockets which grow/are more common with hydrogen content.

TAKEAWAYS POWER INCREASE

The analysis of the power level effects on flame dynamics, stability, and emissions reveals several critical changes.

Flame Dynamics and Stability:

- **Average Measurements:**
At higher power settings (12 kW), hydrogen enrichment results in a more compact flame moving towards the premixing tube, similar to the lower power setting (8 kW). The swirling flame characteristics are maintained, but pure hydrogen conditions disrupt the swirling flow, turning it into a jet-like flow. The higher axial air injection (AAI) percentage necessary to prevent flashback creates a W-shaped flow field compared to the V-shaped flow at lower power.
- **Acoustic and Temporal Flow Measurements:**
The acoustic spectrum at higher power settings shows varying thermoacoustic peaks depending on the hydrogen enrichment level, with significant differences from the low-power mode. These are also present in the flow field. The differences in acoustic responses are influenced by the power setting, flame size, location, AAI, and increased air temperature in the combustion chamber.

Emissions:

- **NO_x Emissions:**
At higher power settings, NO emissions increase significantly compared to the low power mode. The increase is more pronounced at 60% hydrogen enrichment due to higher peak combustion temperatures and the formation of rich pockets within the flame. The thermal NO formation pathway and the NNH pathway contribute to this rise in emissions.
- **CO₂ Emissions:**
The relative CO₂ emissions remain consistent across different power settings, indicating no change in the engine's combustion efficiency at higher power levels.

In conclusion, increasing the power setting to 12 kW amplifies the effects observed with hydrogen enrichment, affecting flame dynamics, acoustics, and emissions. The higher power setting leads to more pronounced changes in flame behavior, acoustics and higher NO emissions.

6.2. EFFECTS OF OSCILLATIONS

In this section the effects of the forced equivalence ratio fluctuations will be discussed. The characterisation of the valve and airflow was already presented in Sec. 4.3, the effects on the general operating conditions, i.e. equivalence ratio, flame speed, ect., were presented in Subsec. 5.1.3. As for the previous effects of enrichment and power the effect of the forced equivalence ratio fluctuations will be discussed first for the average flow field in Subsec. 6.2.1. Then the effect of the modulation on the stability, more specifically the thermoacoustics will be covered in Subsec. 6.2.2. Lastly, the effect on emissions is presented and analysed in Subsec. 6.2.3.

6.2.1. EFFECT ON AVERAGE FLOW FIELD

The forced equivalence ratio fluctuations, inflicted by the actuation of the valve, do not alter the overall shape of the average flow fields inside the combustor as can be seen in Fig. 6.21 and 6.22. The characteristic features like the size of the recirculation zones does not differ from the unmodulated points. This is true across all different power and hydrogen enrichment levels. This is within expectations as the modulation is sinusoidal in nature and symmetric, therefore it is not expected to influence the shape of the average flow field.

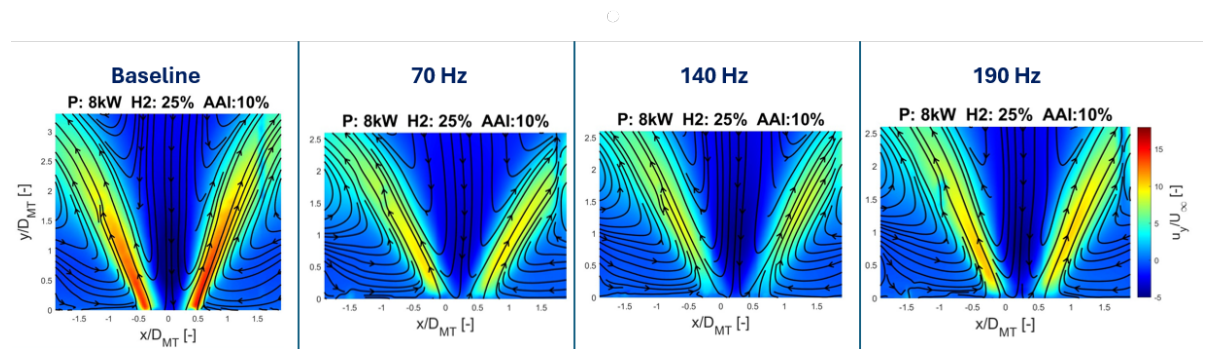


Figure 6.21: Comparison of average velocity field for different actuation frequencies (P: 8 kW, H2: 25 %, AAI: 10%)

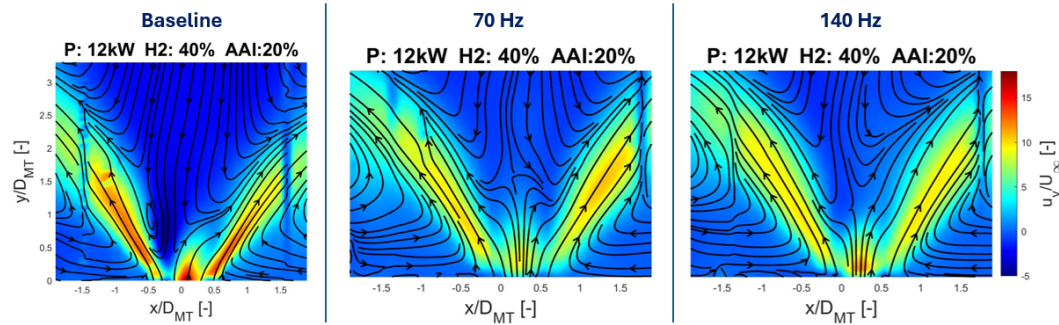


Figure 6.22: Comparison of average velocity field for different actuation frequencies (P: 12 kW, H2: 40%, AAI: 20%)

When looking at the velocities one does spot a decrease in overall velocity for the actuated cases. This can simply be attributed to the reduced mass flow that stems from the actuation of the valve. More precisely with the valve closed 50% of the time of its operations it will limit the overall mass flow as the mass flow in the main air supply line is kept constant. This will lower the overall mass flow and hence speed of the flow according to the continuity or mass flow equation. Given that the mass flow fluctuation is 25 lpm it is expected that the speed roughly decreases changes with same relative amount³.

³So 15% for low power and 10% for high power

6.2.2. EFFECT ON ACOUSTICS AND COHERENT FLOW STRUCTURES

The effect of the valve actuation becomes immediately clear when looking at the external and internal acoustic and pressure measurement presented in Fig. 6.23. It is clear that the noise of the thermoacoustic peak is significantly dampened. This means that the forced fluctuations of the equivalence ratio disrupt the feedback loop of the natural thermoacoustic instability that occurs at this setpoint as presented in Subsec. 6.1.1. The damping occurs for each actuation frequency.

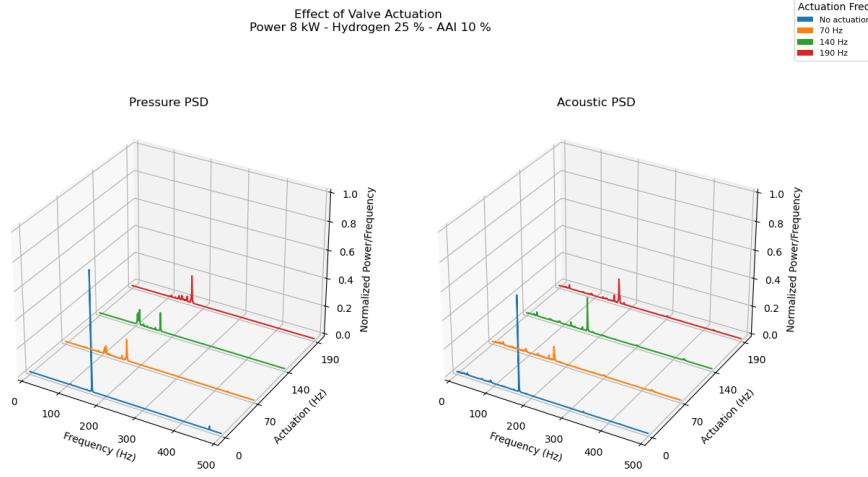


Figure 6.23: Effect of different actuation frequencies on acoustics
Chamber pressure PSD (left) and far-field acoustics (right)
(P:8 kW, H₂: 25 %, AAI:10%)

A possible explanation might be found in the velocity fields. When looking at the single frame velocity fields of any actuated measurement, as shown in Fig. 6.24, a pattern in sync with the actuation frequency will be noticeable. When analysing the flow with the SPOD an energy peak at the actuation frequency is revealed in the energy spectrum, as illustrated in Fig. 6.25. Furthermore, it is clear that the original peak at the unmodulated thermoacoustic instability frequency disappears or becomes lower. This was also noticed by several other studies, presented earlier in Sec. 2.5, which also investigated forced fluctuations in the flow or the equivalence ratio [101, 146].

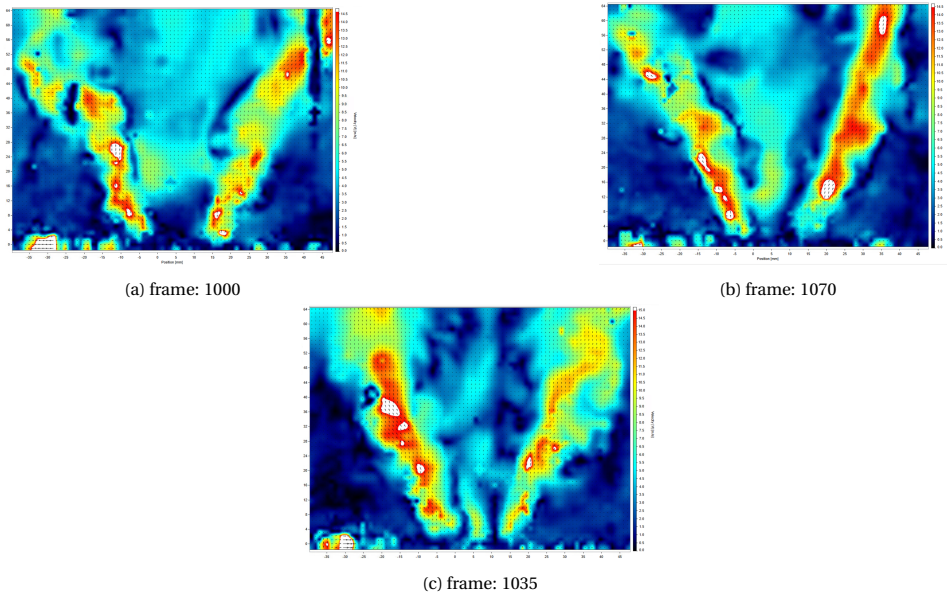


Figure 6.24: Velocity vector field visualised at 3 different time steps in line with actuation of 70 Hz
(P:8 kW, H₂: 25 %, AAI:10%)

The findings by Hashim *et al.* [101] are particularly relevant, and could shed some light on why these forced fluctuations cause the main (thermoacoustic) modes to be dampened. In their study they noted that the forced actuation will create co-winding counter rotating helical vortices, which are located at the beginning of the reverse flow region and the shear layer zone as presented earlier in the literature study. For this research, similar anti-symmetric structures are also present at the underlying flow structures of the peaks in the SPOD energy spectrum. These underlying structures are shown for the different actuation frequencies of the reference base line setpoint in Fig. 6.26. Hashim *et al.* [101] also attributes an oscillatory shift of the stagnation point in axial direction in line with the actuation for their research. This could not be verified for the experiments of this thesis as the OH $\cdot$ -chemiluminescence measurements used to capture the flame location are only low speed and unable to capture the fluctuations at the higher frequencies used. However, if true this could already form an explanation of why the thermoacoustics are dampened by the modulation. Namely, in the previous section, Subsec. 2.5.2.1, it was already noted that the changes in flame location will change the acoustic response of the system. Therefore, the change in stagnation point, and as result the flame location, will affect the feedback loop of the original main response mode as the fluctuation will move the systems response away from this mode.

Furthermore, even without this changing flame location one can already establish that multiple physical transformations resulting from the forced equivalence ratio fluctuations will probably to alter the systems' acoustic response in different ways. For starters the equivalence ratio changes the local gas properties (like density and sound speed), thereby affecting the acoustic properties of the chamber. This change can alter the resonance characteristics of the chamber, potentially shifting the frequencies at which instabilities occur or damping existing modes. Additionally, by introducing equivalence ratio fluctuations, one might also induce a phase shift between the pressure oscillations and the heat release fluctuations. Moreover, Forced fluctuations in the equivalence ratio can alter the heat release rate of the flame. Since thermoacoustic instabilities are driven by the coupling between heat release and acoustic oscillations, changing the timing and magnitude of the heat release can disrupt this coupling, thereby damping the instabilities.

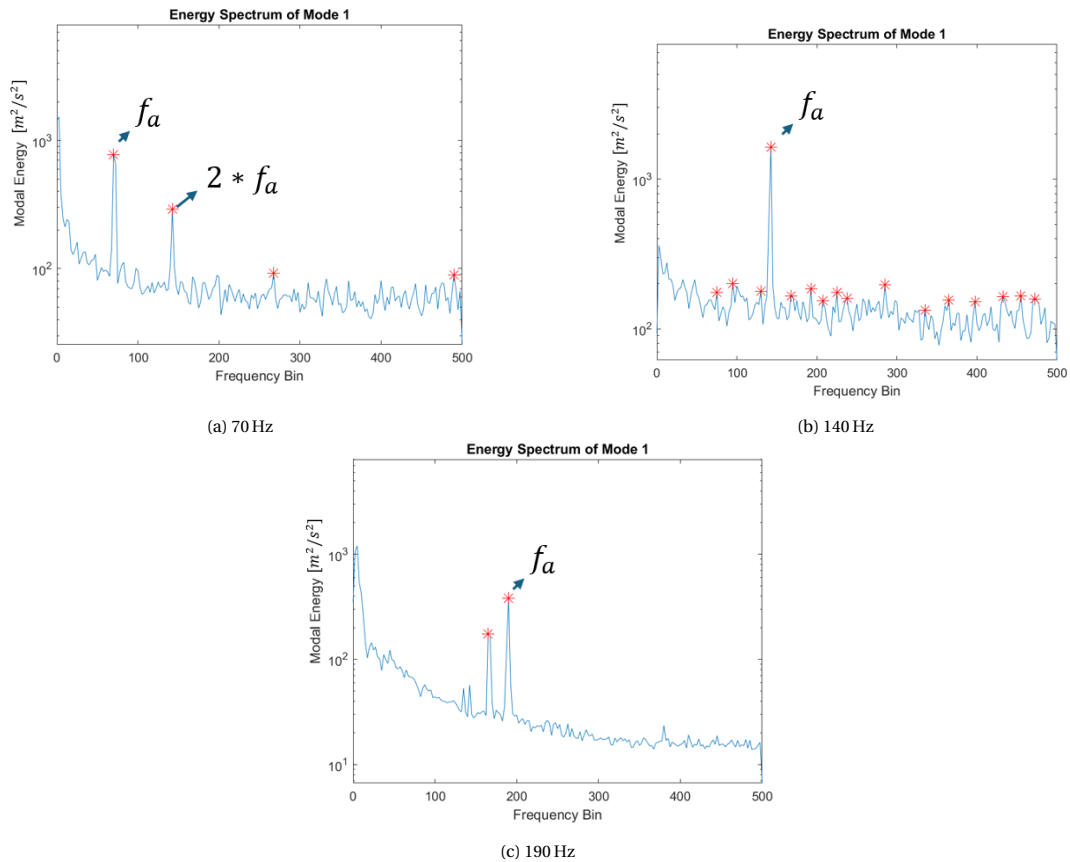


Figure 6.25: Effect of different actuation frequencies first energy spectrum modes of the flow field obtained using SPOD (P:8 kW, H2: 25 %, AAI:10%)

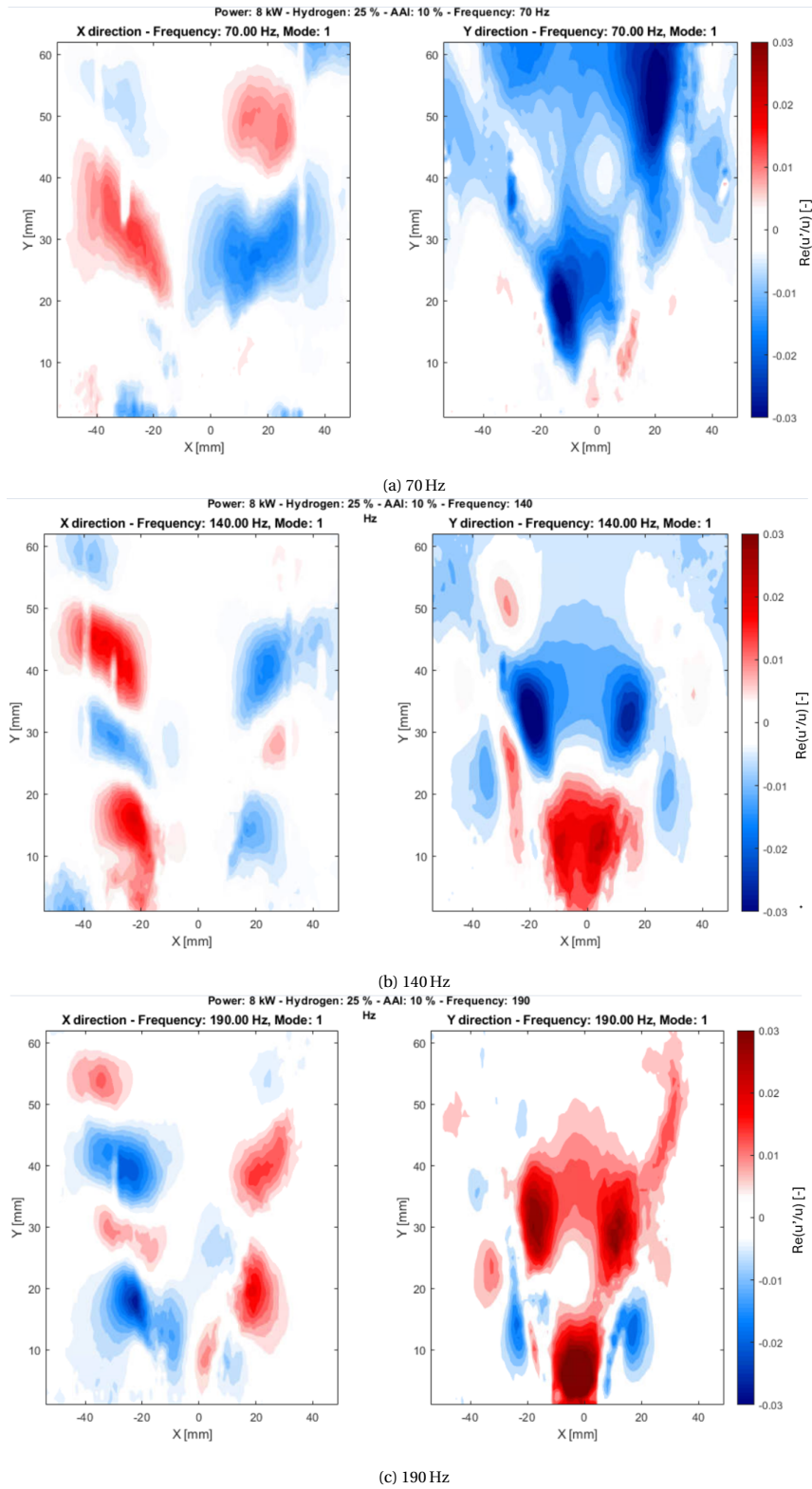


Figure 6.26: Coherent flow structures in the flow field formed at the actuation frequency
 Radial velocity (left) and axial velocity (right) fluctuations at their respective actuation frequency
 (P:8 kW, H2: 25 %, AAI:10%)

EFFECTS FOR HYDROGEN ENRICHMENT

When hydrogen content is increased the effects of the valve actuation remain the same as can be seen in Fig. 6.27 and 6.28. The former also possesses a strong thermoacoustic instability for its actuated base case which also gets suppressed regardless of the actuating frequency. The latter does not possess any natural thermoacoustic instabilities and hence possesses no peaks in its acoustic energy spectrum. Even in this quiet scenario the valve does not produce any noise that is picked up in either the microphone or pressure transducer.

The different energy spectra for the different levels of enrichment and actuation frequency are presented in Fig. 6.29. It is clear that in every spectra the same peaks stemming from the actuation, and their harmonics are present. Indicating that the phenomenon is independent of hydrogen enrichment level.

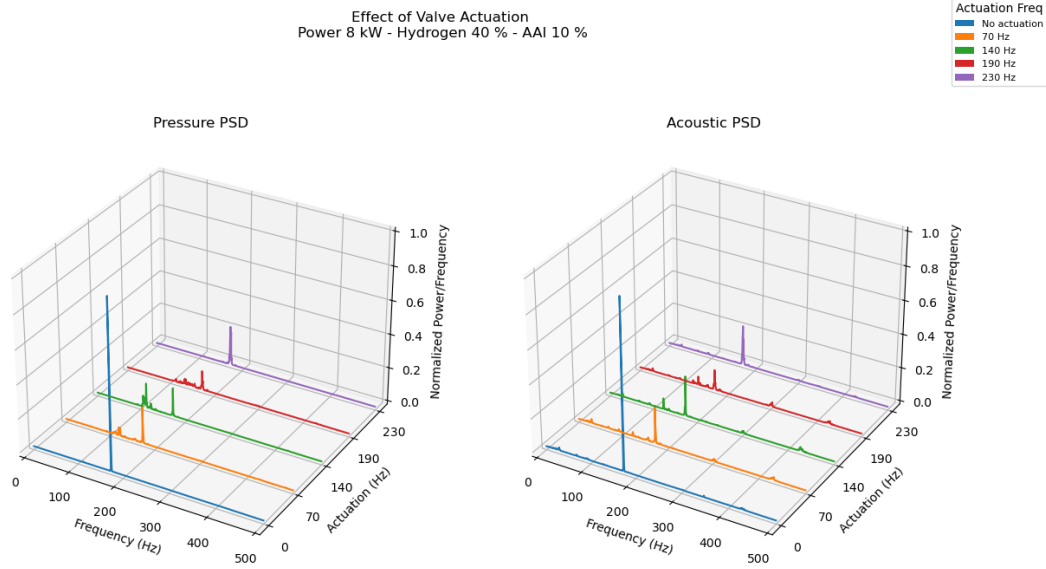


Figure 6.27: Effect of different actuation frequencies on acoustics
Chamber pressure PSD (left) and far-field acoustics (right)
(P:8 kW, H₂: 40 %, AAI:10%)

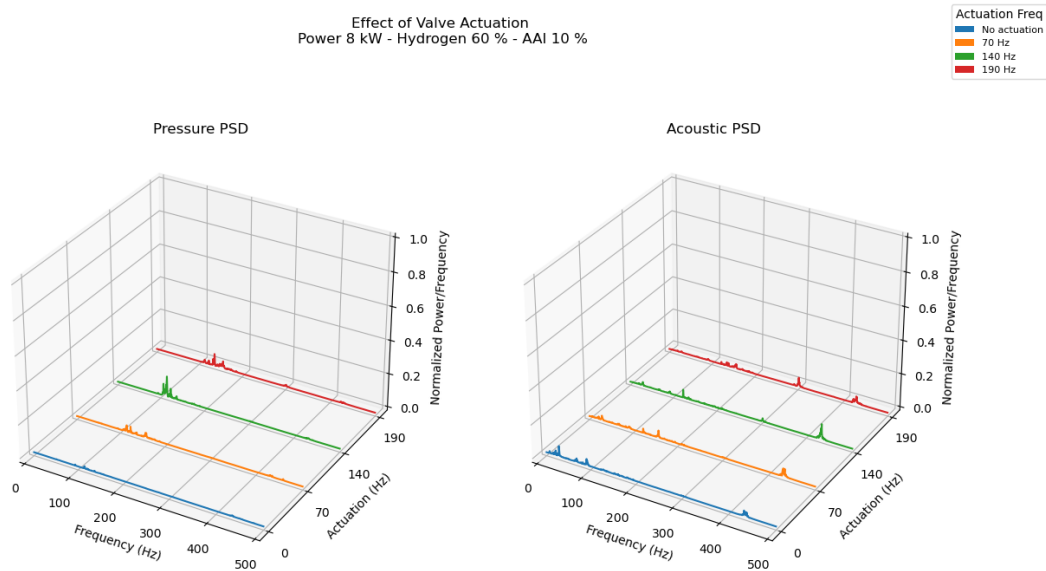


Figure 6.28: Effect of different actuation frequencies on acoustics
Chamber pressure PSD (left) and far-field acoustics (right)
(P:8 kW, H₂: 60 %, AAI:10%)

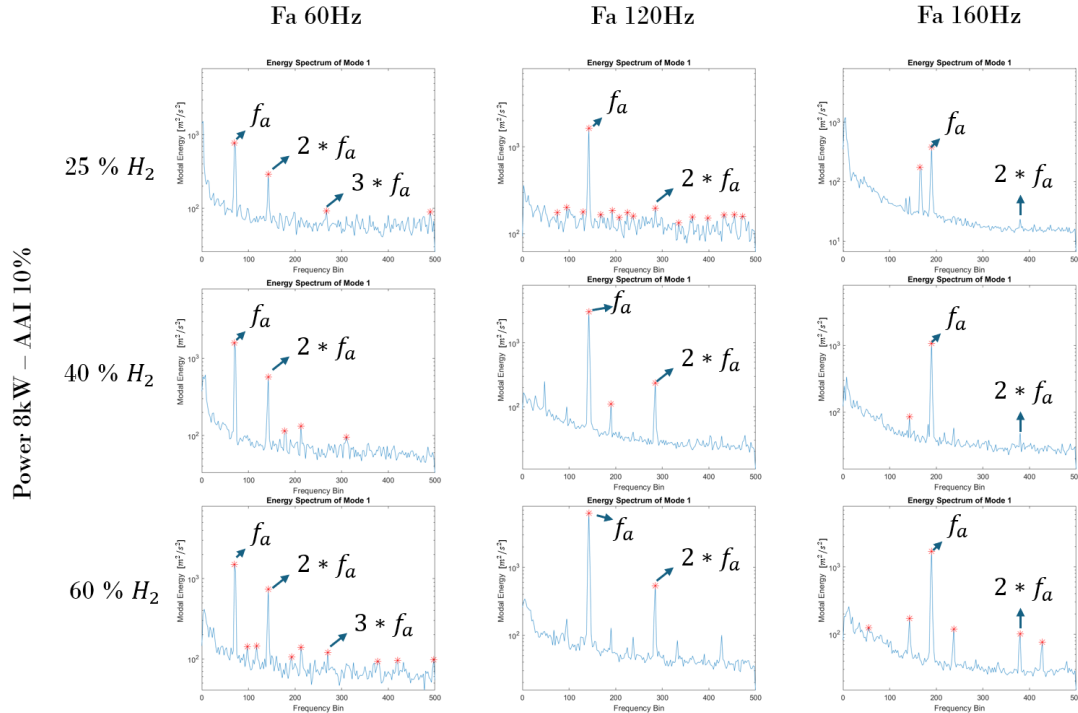


Figure 6.29: Effect of different actuation frequencies on first energy mode spectrum in high power setting (P:12 kW, H2: 0 - 100%, AAI:20%)

EFFECTS FOR POWER LEVEL

In the high power setting the same behaviour with respect to the damping of thermoacoustic instabilities as a result of actuation is measured. For illustration purposes the acoustic power density spectrum's of the 60% hydrogen enriched, high-powered set point has been plotted in Fig. 6.30. The other acoustic spectrum under actuation can be found in the appendix Appendix F.

Once again peaks occur at the valve actuation frequency in the energy spectrum of the flow field obtain through the SPOD. The coherent structure associated with this peaks occur also across hydrogen level in this high-power setting as can be seen in Fig. 6.31 and 6.32.

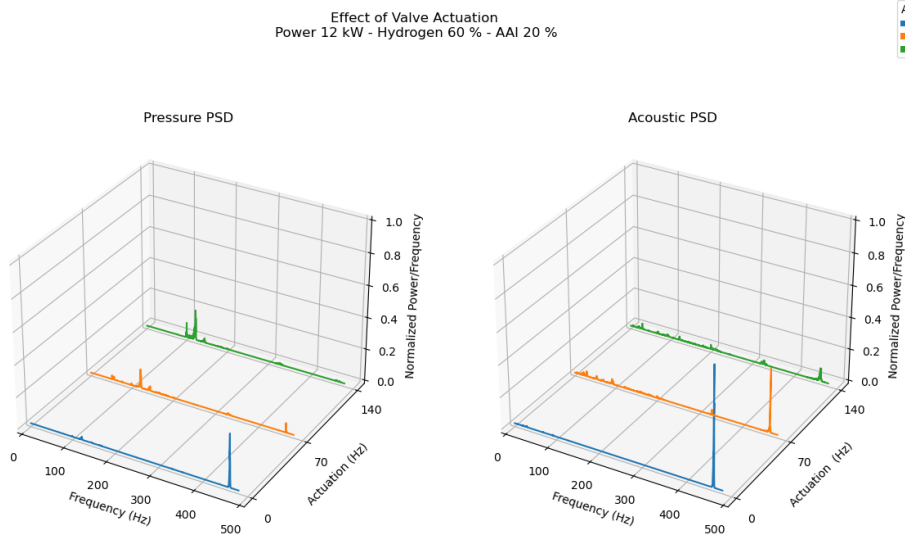


Figure 6.30: Reference case effect of different actuation frequencies on acoustics at higher power setting Chamber pressure PSD (left) and far-field acoustics (right) (P:12 kW, H2: 60 %, AAI:20%)

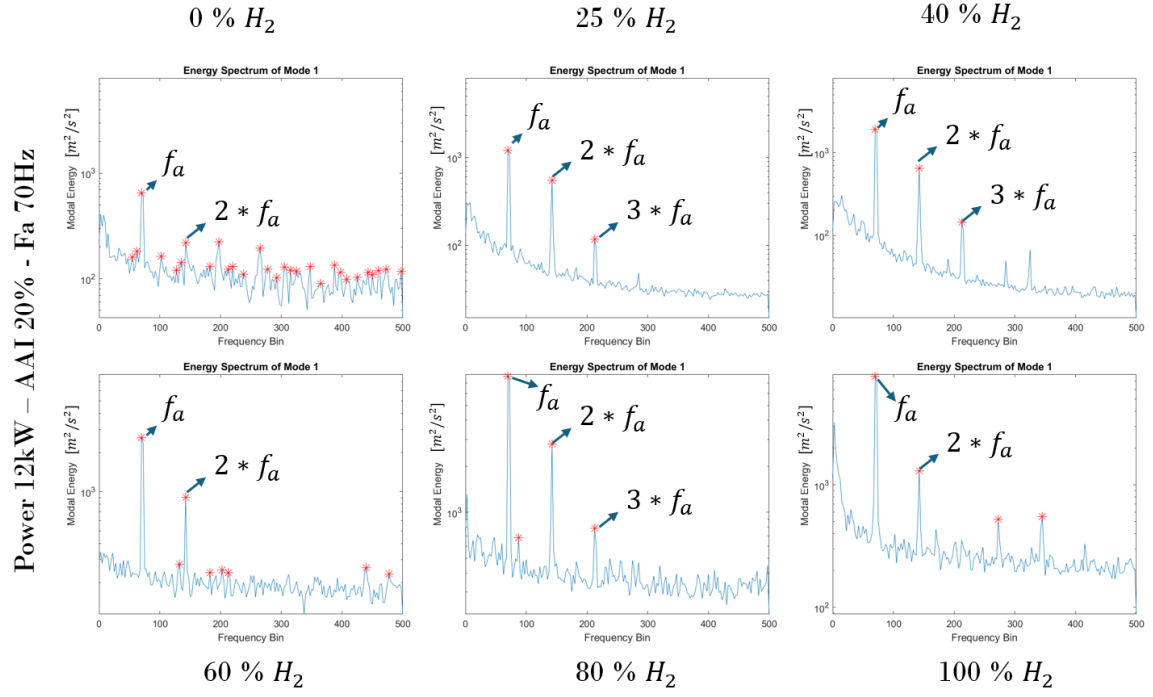


Figure 6.31: Effect of 70 Hz actuation frequencies on first energy mode spectrum in high power setting (P:12 kW, H₂: 0 - 100%, AAI:20%)

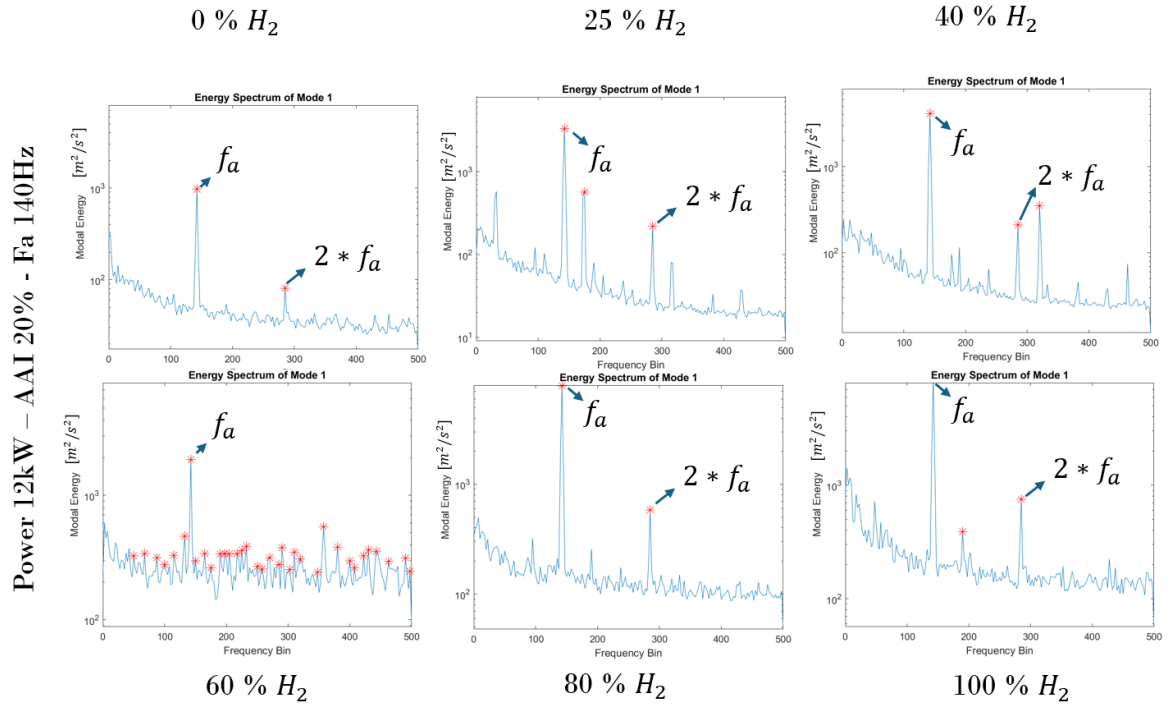


Figure 6.32: Effect of 140 Hz actuation frequencies on first energy mode spectrum in high power setting (P:12 kW, H₂: 0 - 100%, AAI:20%)

6.2.3. EFFECT ON EMISSIONS

The steady state emission measurement collected using the gas analyser showed notable effects of the modulation on both the NO and CO₂ were witnessed. The effect on NO emissions is discussed in Subsec. 6.2.3.1 and the effect on CO₂ in Subsec. 6.2.3.2, respectively.

EFFECT ON NO EMISSIONS

When looking simply at the different NO_x emission levels in Fig. 6.33 for the lower power setting, one sees a clear increase in emissions for all modulation frequencies for the lower enriched points. However, the trend is different for the actuated points, with the 60% enriched point being a lot closer to its baseline value than for the 25% hydrogen enriched point. This increase can be explained by looking into the effects of the valve. Namely, with the valve actuating the actual total air flow is limited not only instantaneously but also on average. The effects of this are twofold. The lower average airflow would mean that if the NO production would remain the same the value would still increase as there will be less air particles in every sample that is taken. This is supported by the lower O₂ readings in the actuated points, e.g. for the reference baseline point (Power: 8 kW, Hydrogen: 25% this was 5% lower. Furthermore, as established in Subsec. 5.1.3, more precisely Table 5.4 and 5.5, lower airflow would actually mean that the mixture is richer, and thus hotter, resulting in more thermal NO_x production as seen in Fig. 6.33a. However, in Fig. 6.33b one sees that this is not true for the higher hydrogen enriched points.

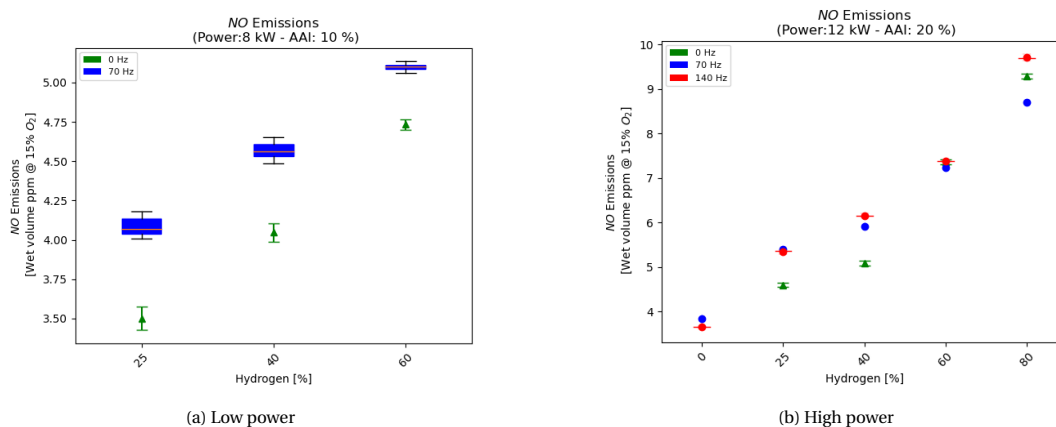


Figure 6.33: Effect of actuation on NO emissions in different power settings for different hydrogen enrichment levels.

In Fig. 6.33b one observes once again that emissions are higher for the actuated setpoints than the unmodulated points for the points with low hydrogen enrichment. However, for points with higher hydrogen content, the emissions are equal to or lower than expected. This is particularly notable given that the air content in each sample is also reduced. It is well established that equivalence ratios significantly influence NO_x production pathways for the same enrichment level. For higher equivalence ratios, as seen in the 25% and 40% hydrogen content points, thermal NO_x is the dominant pathway as the temperatures are significantly higher than the highly enriched hydrogen points as indicated by Table 5.4 and ???. Conversely, at lower equivalence ratios, such as 60% and 80%, other pathways like the NNH route become more dominant. This difference in NO_x production likely underlies the varying measurements. For instance, the NNH pathway is known to be promoted by locally rich conditions⁴. The improved mixing caused by the difference in turbulence⁵, pressure and velocity from actuation could reduce these locally rich conditions, leading to less NNH NO production. However, further research is necessary to confirm this hypothesis.

⁴Which are suggested to be present according to preliminary numerical simulations

⁵With the introduction of new coherent structures as established in the previous section

EFFECT ON CO₂ EMISSIONS

The modulations also increase the CO₂ emissions in both the low- and high-power cases as seen in Fig. 6.34. However, the downward trend with hydrogen enrichment remains intact. This can more straight forwardly just be attributed to the richer conditions, caused by the flow limit imposed by the valve, in the actuated cases. Since the total airflow is lowered and the fuel content remains the same, the relative volumetric percentage of the emissions increases as expected.

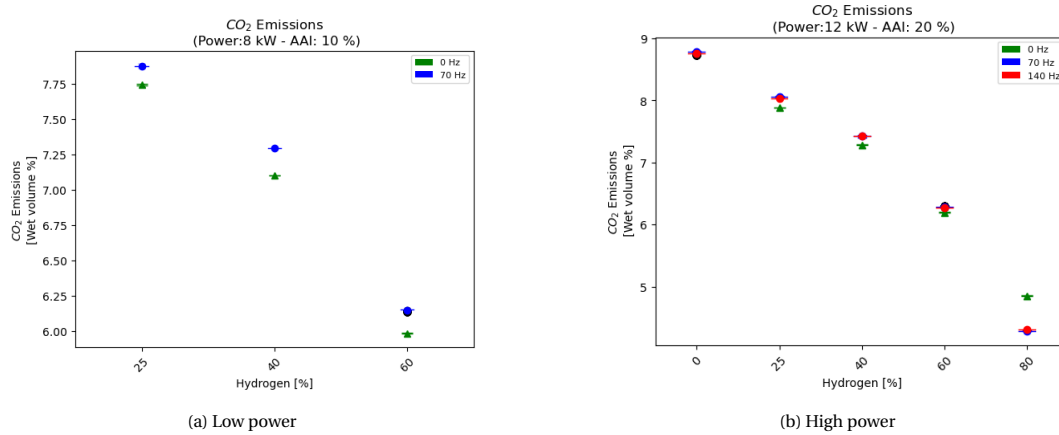


Figure 6.34: Effect of actuation on CO₂ emissions in different power settings for different hydrogen enrichment levels.

TAKEAWAYS FORCED EQUIVALENCE RATIO FLUCTUATIONS

The forced equivalence ratio fluctuations induced by valve actuation have a significant effect combustion dynamics. These fluctuations impact various aspects of the combustion process, including the average flow field, acoustic characteristics, and emission profiles. The following points summarise the key findings from the analysis of these effects.

Flame Dynamics and Stability:

- **Average Measurements:**
The forced equivalence ratio fluctuations do not significantly alter the overall shape of the average flow fields. The characteristic features, such as recirculation zones, remain consistent across different power and hydrogen enrichment levels. However, a decrease in overall velocity is noted due to the reduced mass flow from the valve actuation.
- **Acoustic and Temporal Flow Measurements:**
The forced fluctuations significantly dampen the thermoacoustic peaks in both low and high power settings. This disruption in the natural thermoacoustic instability feedback loop occurs across all actuation frequencies. The introduction of forced fluctuations generates co-winding counter-rotating helical vortices, affecting the system's acoustic response by changing local gas properties and inducing phase shifts between pressure oscillations and heat release fluctuations. The energy spectrum of the flow field reveals peaks at the actuation frequencies, indicating the presence of coherent structures linked to these frequencies. These structures remain consistent across different hydrogen enrichment levels and power settings.

Emissions:

- **NO_x Emissions:**
The forced equivalence ratio fluctuations generally increase NO emissions, especially at lower hydrogen enrichment levels. The reduced average airflow due to actuation results in richer conditions and higher thermal NO production. However, at higher hydrogen enrichment levels, the NO emissions trend shows a different behavior, potentially due to changes in the dominant NO production pathways and improved mixing reducing locally rich conditions.
- **CO₂ Emissions:**
The CO₂ emissions increase with forced fluctuations in both low- and high-power settings, maintaining the downward trend with increasing hydrogen enrichment. This increase is attributed to the richer conditions from reduced airflow.

In conclusion, forced equivalence ratio fluctuations have a significant impact on the flame dynamics, acoustics, and emissions. While they effectively dampen thermoacoustic instabilities and create distinct coherent flow structures, they also lead to increased emissions due to richer combustion conditions.

7

NUMERICAL ANALYSIS COMBUSTION INSTABILITIES

In order to verify the claims made in the previous chapter and expand upon the findings by simulating experimental conditions that are closer to the operations of real jet engines a numerical analysis was conducted. For this Oscilos¹, an open source combustion instability low order simulator developed at imperial college, is used. This low order simulator based in Matlab allows for the simulation of both longitudinal and annular combustor geometries by modelling them as a network of connected modules. It can simulate acoustic waves for a wide variety of flame and boundary conditions.

In this chapter the low order numerical analysis of the combustor combustion instabilities is introduced and the findings are discussed. The digital model of the experimental setup is detailed in Sec. 7.1, followed by a comparison of the combustion instabilities witnessed during the experiments and those obtained by the numerical model in Sec. 7.2. The effects of the flame location, which was seen to change a lot during the experiments, is analysed in Sec. 7.3. Then, the effect of the chamber geometry on the combustion instabilities is discussed in Sec. 7.4. The next section, Sec. 7.5, goes one step further and assess the impact of the operating conditions on the natural combustion instabilities. Lastly, the effect of varying or forced inflow conditions, similar to the fluctuations of the forced equivalence ratio, is discussed in Sec. 7.6.

7.1. BASELINE MODEL OF EXPERIMENTAL COMBUSTOR

Oscilos allows one to easily model and analyse the combustion instabilities of a setup at the hand of the specifications of a few key building blocks. These necessary specifications pertain to the combustor geometry, thermal properties and boundary conditions. Additionally, a flame model needs to be supplied or chosen.

7.1.1. GEOMETRY

One can construct a two-dimensional model of their own chamber geometry in Oscilos as seen in Fig. 7.1. The dimensions shown are measured and taken from the actual experimental setup that was used. However, due to limitations in the software the flexible exhaust tube had to be assumed straight². Furthermore, one has to specify the location of the flame in the chamber; this was assumed to be at the exit of the premixing tube.

¹<https://oscilos.com/index.html>

²This is deemed acceptable [183] as the basic resonance frequency formula for a tube is determined by the length (L) and speed of sound (ν) of the surrounding air. $f = \frac{n\nu}{2L}$

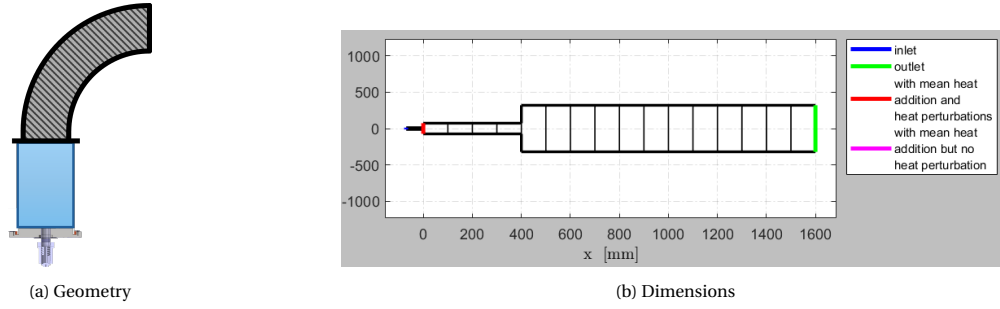


Figure 7.1: Experimental chamber geometry used numerical analysis

7.1.2. THERMAL PROPERTIES

Besides the chamber dimensions and flame location, inlet mean flow properties and heat addition style also need to be provided. The mean flow inlet properties were set to the standard atmospheric conditions as the experimental chamber is experimental. The bulk axial flow velocity was calculated to be around 8.75 ms^{-1} . The resulting mean temperature throughout the chamber is given in Fig. 7.2.

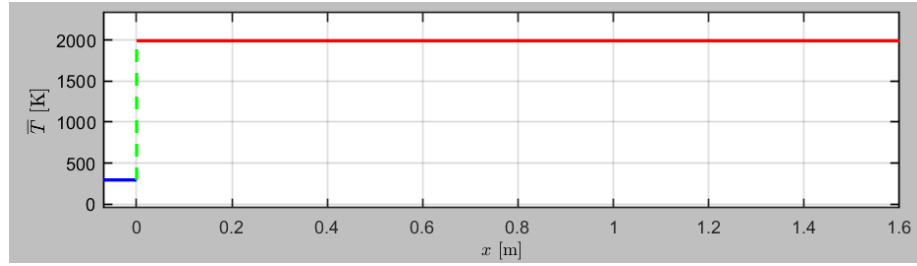


Figure 7.2: Mean temperature in combustor

The heat addition style was chosen to be heating grid, for which the temperature ratio across the grid (T_b/T_u) needs to be specified. This was calculated using Cantera³ for a 40% hydrogen enriched methane flame with an equivalence ratio of 0.766 matching one of the setpoints obtained during the campaign. The calculated ratio between the upstream and down stream temperatures came down to 6.897.

7.1.3. FLAME MODEL

The unsteady heat sources was initially modelled using a linear flame transfer function model provided by Oscilos. The linear flame model is based on Crocco's $n - \tau$ model [184] filtered by a second order filter. The transfer function $\mathcal{T}_u$, shown in Eq. 7.1, is defined by the Laplace variable s , cut-off frequency $f_c = \omega_c/2\pi$, gain a_f , time delay τ_f and dampening ratio ξ .

$$\mathcal{T}_u(s) = \frac{\omega_c^2}{s^2 + 2\xi\omega_c s + \omega_c^2} a_f e^{-\tau_f s} \quad (7.1)$$

The standard values of $a_f = 1$, $f_c = 75 \text{ Hz}$, $\tau_f = 3 \text{ ms}$, and $\xi = 1$ were kept for the model.

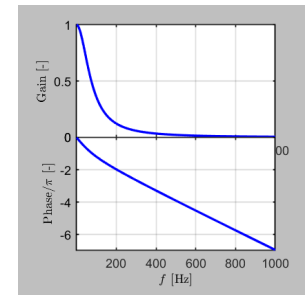


Figure 7.3: Linear flame FTF

³<https://cantera.org/>

7.1.4. BOUNDARY CONDITIONS

Lastly, to complete the model the inlet and outlet boundary conditions need to be specified. Oscilos assumes the sidewalls to be adiabatic. This is not a realistic assumption for the lab-scale or real-scale experiments, where heat is transferred to the surroundings. This will increase the temperature, and therefore speed of sound and thus natural frequency, in the model. For the reference case, these boundary conditions are quite straightforward. A closed condition needs to be set on the inlet and an open on the atmospheric outlet. The transfer function for both can be seen in Fig. 7.4. The reflection coefficient for the inlet is 1 and for the outlet -1 . This means that the inlet is rigid and does not allow for any displacement. Therefore reflected wave has the same amplitude and phase as the incident wave. At the outlet, the reflected wave has the same amplitude as the incident wave but is inverted (180 degrees out of phase).

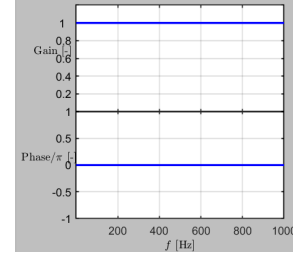


Figure 7.4: Standard boundary condition Inlet

7.2. RESULT OF NUMERICAL MODEL BASELINE

The resulting frequency domain for the 1d model of the experimental set-up used in the campaign can be visualised in an spectral contour plot. The plot for the 40% hydrogen enriched condition is presented in Fig. 7.5. One can clearly see that the natural frequency was measured at around 160 Hz and indicated with a ☆, which was also apparent for the in the experimental measurements. The contour lines indicate $20\log_{10}|\delta e(s)|$ or the logarithmic magnitude of the system's transfer function in the s-plane. This magnitude tells one how much each frequency component of the impulse is amplified or attenuated by the system.

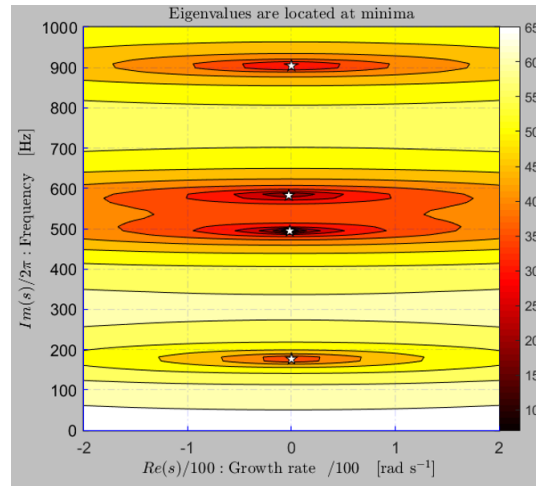


Figure 7.5: Contour map of $20\log_{10}|\delta e(s)|$ in the s-plane for experimental combustion chamber with the main modes of the chamber indicated by a ☆

7.3. EFFECT OF FLAME LOCATION

Within the experiments it was seen that the flame location is susceptible to change depending on the hydrogen enrichment level, as described in Subsec. 6.1.2. This effect can be analysed in the 1d model, by moving the location of the heating grid. Two alternative heat addition locations, located further down the chamber, were simulated. The chosen locations, i.e. $0.05m$ and $0.1m$ respectively, were taken to be comparable to what was witnessed during the experiments and are indicated in Fig. 7.6.

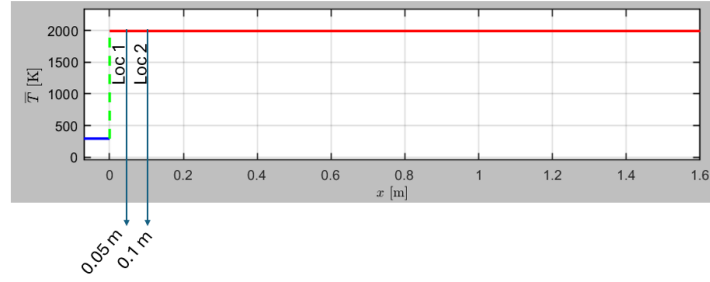
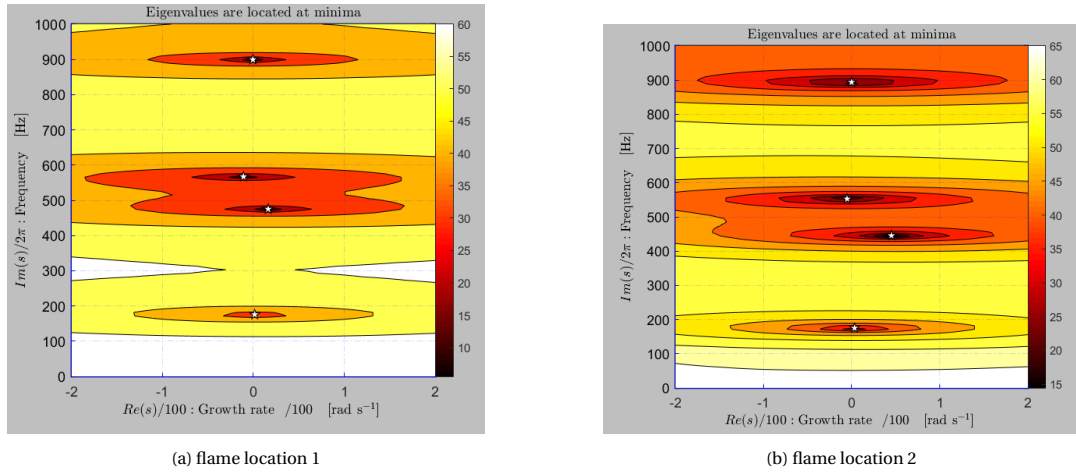


Figure 7.6: Alternative locations for heat additions in 1d model

The different flame location, which can be interpreted as a lifted flame, shows clear changes in the contour map of $20\log_{10}|\delta e(s)|$ in the s -plane presented Fig. 7.7. It is mainly the real part for both the main modes and the impulse response that is affected by the different heat addition locations. The frequency, or imaginary part, of the natural instabilities stays the same. The positive growth rate, for the instability at 450 Hz means that this mode has become unstable and will continue to grow.

Figure 7.7: Contour map of $20\log_{10}|\delta e(s)|$ in the s -plane for different flame locations with the main modes of the chamber indicated by a ☆

7.4. EFFECT OF GEOMETRIC CHANGES ON NATURAL INSTABILITIES

One of the major aspects that make the experimental set-up unrealistic is the large exhaust tube attached to the combustor. This exhaust tube functions as a large trombone and results in very low natural acoustic frequencies as was made evidently clear from the experimental measurements and findings in the previous section. Real engines combustion chamber are lot compacter in order to save weight, which is one of the key factor for aero-engine combustor design. In order to get an idea of the impact this change in combustor length has on the design, the exhaust was removed of the model.

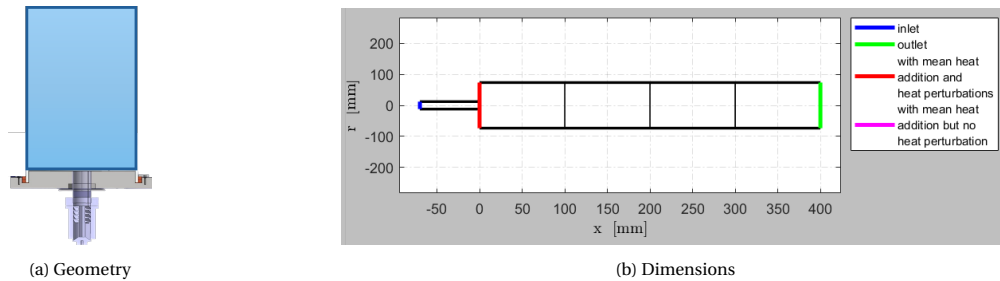


Figure 7.8: Shortened chamber geometry used numerical analysis

Even with just this small alteration the impact on the frequency domain of the combustion instabilities is

made abundantly clear when looking at Fig. 7.8. The smaller geometry will increase the frequency of the instabilities significantly. An even smaller geometry would push this value even higher.

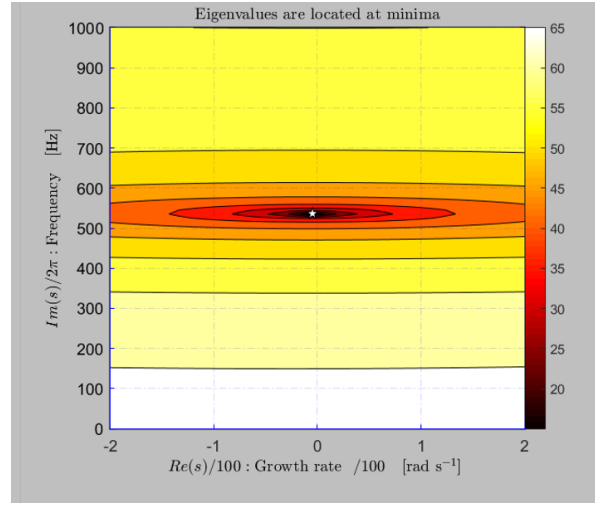


Figure 7.9: Contour map of $20\log_{10}|\delta e(s)|$ in the s -plane for shortened combustion chamber with the main modes of the chamber indicated by a ☆

7.5. EFFECT OF OPERATIONAL CHANGES ON NATURAL INSTABILITIES

In addition to the geometry, the operating conditions also have a large impact on the frequency of the combustion instabilities due to the difference in speed of sound. Normal jet engine combustion chambers operate at high pressure and temperature in order to achieve more efficient combustion. By increasing the thermal properties in the Oscillos model, one can simulate what the effect of these more challenging operating conditions would be on the combustion instabilities.

In order to model these more realistic operating conditions it was decided to increase the pressure to 10 bar and the inlet temperature to 700 K. Furthermore, the flame temperature ratio was decreased in order to keep the peak temperatures within the combustor realistic, i.e. around 2500 K. In this case the power increases from 12 kW to 37 kW. Additionally, the outlet boundary condition needs to be changed to a choked condition to simulate a turbine inlet which are typically choked for flow control.

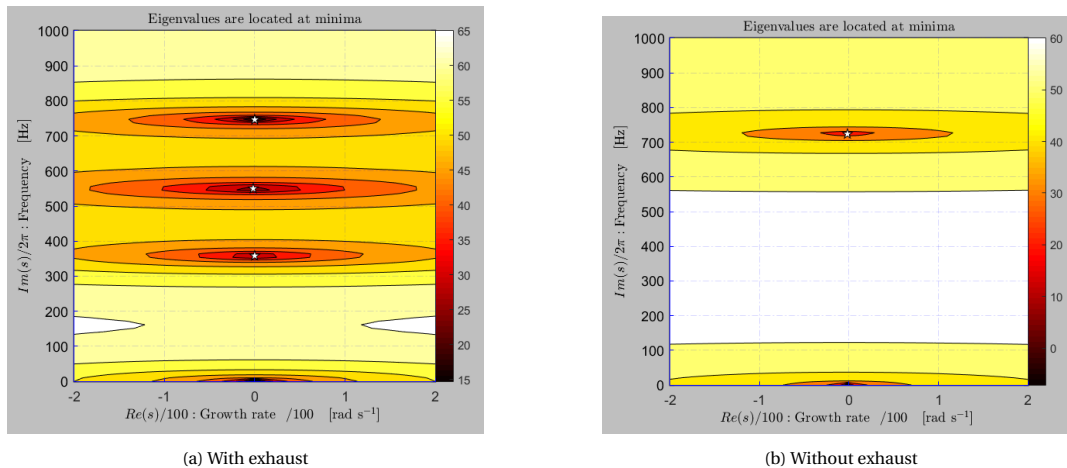


Figure 7.10: Contour map of $20\log_{10}|\delta e(s)|$ in the s -plane for increased operating conditions with the main modes of the chamber indicated by a ☆

The resulting frequency domains of the eigenvalues, shown in Fig. 7.10, tell a similar story as the effect of the geometry. The natural frequencies are seen to increase regardless of the chamber geometry, due to the increase in density and temperature that will increase the speed of sound and therefore the natural frequency

of the tube. The growth rate remains unchanged, as indicated by the locations of the eigenvalues with respect to the x-axis and the contour plot scales.

7.6. EFFECT OF FLUCTUATIONS ON NATURAL INSTABILITIES

The higher natural frequency of combustion instabilities, whether due to a shorter combustor geometry or increased operating conditions, raises the question of whether the dampening effect of forced equivalence ratio fluctuations, as noted in experimental research, would also be applicable at these higher frequencies. This can be further explored using Oscillos, which allows for modelling velocity perturbations ($u'/\bar{u}$) in the context of a non-linear flame function.

To address this, the linear flame model for the Flame Transfer Function (FTF) is first transformed into a non-linear model. This model comprises both linear and non-linear components, with the non-linear component based on the model by [Li and Morgans \[185\]](#). The unsteady velocity ratio is set to 10% of the total velocity. Unfortunately, it is not possible to attribute a frequency to this disturbance.

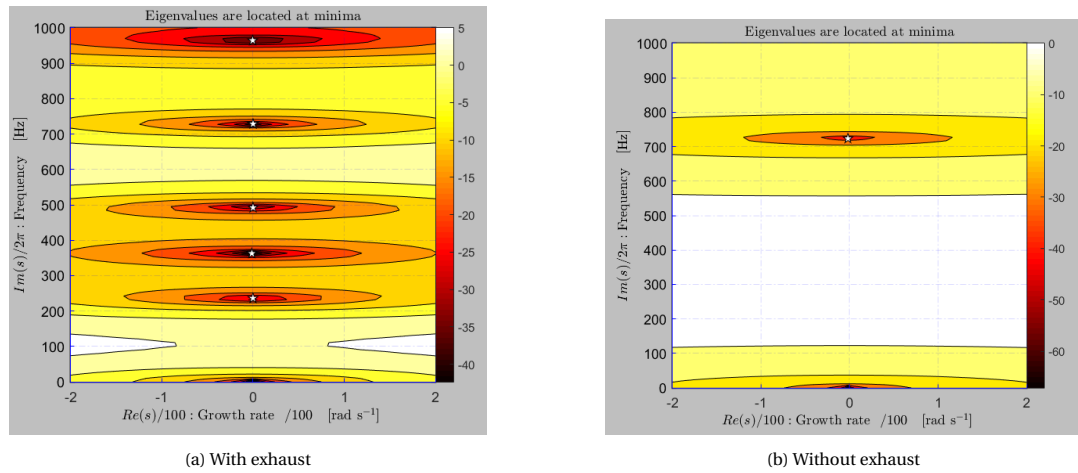


Figure 7.11: Contour map of $20\log_{10}|\delta e(s)|$ in the s-plane for velocity perturbations with the main modes of the chamber indicated by a ☆

A similar effect is seen in both contour plots presented in [Fig. 7.11](#). Namely, the magnitude of the impulse response growth rates, indicated by the contour lines, become significantly more negative. Indicating, that perturbations around the eigenvalues will decay instead of grow. Additionally, in [Fig. 7.11a](#) one can see the addition of several new natural combustion instability frequencies. Most likely, the smaller combustion chamber also develops additional. Therefore, because the growth becomes negative, indicating a decay, the indication would be that the perturbation will cause the natural combustion frequencies to be dampened as these eigenvalues' growth rates are close to zero but the surrounding frequency spectrum becomes negative. This will cause impulses that would normally reinforce these combustion instabilities to decay quicker.

To further reinforce this statement, it was also observed in the experimental campaign that the large frequency difference, e.g. 70 Hz and 450 Hz in [Subsec. 6.2.2.1](#), between the modulation frequency and the frequency of the natural combustion instability did not hamper the ability of the modulation to dampen the combustion instability.

8

CONCLUSION

Harder-to-abate industries, such as the aviation industry, need to look into alternatives for their carbon-based fuels, or they will eventually be forced to change by regulations or the consequences of climate change itself. In the context of gasturbines and/or jet engines electrification is often not an option, hence alternative fuels, like hydrogen or biofuels, will need to be used. However, due to the limited availability and steep prices, the first new engine designs need to be multi-fuel capable to allow for the enrichment of conventional fuels.

The TU Delft Sustainable Propulsion Lab is currently advancing research on new combustion technologies through its APPU project, specifically focusing on a lean, premixed, swirl-stabilised methane combustor that can be adapted for full hydrogen capability. One of the major areas of development is the control and understanding of the engine's thermoacoustic instabilities. These instabilities that are accompanied with high damaging sound levels can have many origins and are determined by many parameters. One of the possible origins are equivalence fluctuations that stem from airflow fluctuations. This topic, especially in combination with hydrogen enrichment, has been researched very little thus far in the broader scientific community. This master thesis looked into expanding the knowledge on this topic by researching the effects of forced equivalence ratio fluctuations on the flame dynamics, stability and emissions.

These forced equivalence ratio fluctuations were achieved by the installation of a high speed solenoid flow valve on a bypass air line added to the main air of the setup. This allowed for the modulation of said airflow in a precise and controlled manner. This forced airflow actuation was then applied several unique set points that span the operational range in both hydrogen content and power level. These set-points were then analysed using multiple different measurement techniques focusing on different aspects of the flame. High-speed Particle Image Velocimetry (PIV) was used to study the flow field. Additional SPOD analysis on these instantaneous velocity data gained temporal insights on the governing underlying coherent structures within the flow field. In chamber and external pressure and acoustic measurements helped to gain insights on the thermo-acoustics. Furthermore, emission data was collected using a gas analyser.

Next to the experimental work a low-order numerical model of the system was created in Oscillos, an open source modelling tool for combustion instabilities, in order to translate the experimental findings to real world settings. The model was also used to verify some of the claims made on the experimental findings.

The findings stemming from the experimental measurements and numerical study can best be shared via answering the sub research questions of the research:

To what extend do the imposed equivalence ratio fluctuations alter the flow field of the enriched methane flame? And does the effect change with power level, valve frequency or hydrogen quantity?

The overall shape of the average flow field is not altered by the forced sinusoidal equivalence ratio fluctuations applied in this thesis. Typical characteristics features such as the recirculation zones and the locations of the jets remain intact regardless of power level, valve frequency or hydrogen quantity.

The peak overall velocity within the flow field decreases slightly due to the lower total mass flow as a result from the valve actuating. The valve will be closed and thus limit air flow that is not compensated by the total air supply. This will effectively lower the overall mass flow and consequently the velocity,

Additionally, through the SPOD analysis it was established that the forced fluctuations will generate new coherent flow structures at their respective actuation frequency within the flow field. These newly formed structures are believed to be hydrodynamic instabilities, more specifically dynamic pressure or velocity fluctuations, as they are not captured by either the microphone or in chamber static pressure transducer. These structures are formed regardless of power level, hydrogen content or actuation frequency.

To what extent does the imposed frequency of equivalence ratio fluctuations influence the stability, more specifically the acoustics and pressure waves, of the hydrogen enriched methane flame?

From the experimental results, the application of the valve actuation has a clear impact on the flame dynamics. The frequency-specific damping effect observed can be attributed to several factors: the introduction of co-winding counter-rotating helical vortices at the beginning of the reverse flow region and the shear layer zone, and the possible oscillatory shifts of the stagnation point due to the actuation. These structural changes disrupt the feedback loop essential for sustaining the natural thermoacoustic modes, leading to a reduction in the amplitude of these modes across various actuation frequencies.

Furthermore, the alterations in the equivalence ratio caused by valve actuation influence the local gas properties, such as density and sound speed, which in turn affect the acoustic properties of the combustion chamber. This results in a shift or dampening of the resonance characteristics, leading to a more stable flame condition even at different hydrogen enrichment levels and power settings. The findings indicate that irrespective of the hydrogen content—from the base level to higher enrichment levels—the imposed fluctuations uniformly dampen the peaks associated with thermoacoustic instabilities.

In conclusion, the stability of hydrogen-enriched methane flames is significantly influenced by the frequency of imposed equivalence ratio fluctuations. The uniform suppression of thermoacoustic instabilities across different settings demonstrates a robust interaction between the forced equivalence ratio fluctuations and the inherent acoustic and pressure dynamics of the system. This systematic dampening not only elucidates the mechanical interplay between valve actuation and flame behavior but also highlights the potential of using controlled fluctuations to enhance flame stability in practical applications.

How do the CO₂ and NO emissions of the hydrogen enriched methane flame change when equivalence ratio fluctuations are imposed on the flame?

The CO₂ emissions increase when the valve is actuated due to richer operating conditions originating from the flow restrictions imposed by the valve. The valve closure will limit the airflow and create richer operating conditions. The lower airflow will increase portion of emissions relative to total air flow.

The NO either increase with valve actuation for the same reason at low hydrogen enrichment levels. Or, remain unchanged regardless of the increase in equivalence ratio and associated temperature rise. This can be likely be explained through the equivalence ratio of the different unmodulated set-points and the associated NO_x production pathways. The lower powered and/or low hydrogen enriched set-points have a far higher equivalence ratio than the high hydrogen set-points for which the NNH route is more dominant. The NO_x production for the aforementioned high equivalence ratio points are dominated by thermal NO_x, which is far more susceptible to a small rise in temperature and equivalence ratio than the NNH route for which the improved turbulence and mixing is disadvantageous. However, additional research is needed to confirm this.

How do the findings related to the imposed equivalence ratio fluctuations on hydrogen enriched methane flames in a lab scale premixed swirler combustor translated to larger systems using a low order combustion instability model of the system?

The labscale combustor has a big exhaust tube attached to it, which will function as a big trombone during operations giving way to the very low natural frequencies of the combustion instabilities that were witnessed during the experiments. Decreasing the length of the combustion chamber would increase the natural frequency of the combustion instabilities. Furthermore, the operating conditions are typically elevated in larger systems, which would lead to even higher natural frequencies do to a higher speed of sound. Within the low-order model it was possible to create a shorter, higher powered version of the labscale combustor that operates at elevated temperature and pressure. It was possible to model inlet velocity fluctuations and their effect on the combustion instabilities within the model. The model showed that these velocity perturbations increased the decay rate and that the frequency of the natural combustion instabilities remains unchanged. Although the velocity perturbations are not an exact interpretation of the effects caused by the forced equivalence ratio fluctuations inflicted by a flow valve, they do mimic some of the aspects of these modulations.

9

RECOMMENDATIONS

In this final chapter, the research is reflected upon, and practical suggestions are proposed to improve future experiments related to the topic and within the laboratory. Additionally, ideas for future research will be suggested. Starting with the improvements:

- **Preheating:**

The varying temperatures of inlet air and fuel during experiments have influenced the acoustic response of the combustion chamber, as the speed of sound changes with temperature. By integrating and modifying the available preheating system, this temperature variability can be controlled, ensuring more uniform experimental conditions and more reliable data on the chamber's acoustic properties. Effectively, taking another variable out of the equation.

- **Realistic exhaust:**

As highlighted in the numerical analysis chapter, the large exhaust tube used in lab settings does not accurately represent real engine conditions. To enhance realism and applicability, improvements could be made by installing a plate with a hole between the combustion chamber and the exhaust. This modification would better simulate the exhaust flow dynamics of actual engines, providing more relevant data for practical applications.

- **Additional chamber pressure reading:**

An unused port on the bottom plate of the chamber offers an opportunity to enhance diagnostic capabilities. By installing an additional pressure sensor in this location, it would be possible to more accurately differentiate between the various modes of pressure fluctuations—be they radial, axial, or circumferential. This enhancement would provide a more detailed understanding of the dynamic behaviors within the combustion chamber.

Moving on to future research directions, the following ideas are proposed:

- **PVC actuation:**

With a thorough understanding and characterization of valve control now achieved, it would be intriguing to actuate the flame at the frequency of the Precessing Vortex Core (PVC) observed in cold flows as originally intended in this research. This exploration could determine if a similar PVC develops in reacting flows as is the case for some of the research presented from literature [146], providing deeper insights into flame dynamics.

- **(Numerical) research on NO emissions pathways:**

Further research is needed to verify the hypothesis concerning NO_x emissions. Initial findings suggest NO levels increase at low hydrogen enrichment due to richer conditions or remain unchanged at high hydrogen levels due to different NO_x pathways. Specifically, lower power settings favor thermal NO_x production, sensitive to small changes in conditions, whereas higher hydrogen settings favor the NNH pathway, less affected by these changes. This requires detailed (numerical) analysis to confirm.

- **PLIF Campaign:**

Now that the PLIF (Planar Laser-Induced Fluorescence) laser is available, it presents an excellent opportunity to conduct a detailed study of the flame front. This study would focus on analyzing the concentrations of both OH and CH, for which both the necessary dies were purchased. Such visualization would be invaluable for understanding the effects of dampening on the thermo-acoustic phenomena in the combustion process.

- **Premixing tube study:**

The interaction between the swirling flow and the Axial Air Injection (AAI) in the current setup has not yet been thoroughly investigated. Ongoing numerical simulations indicate that the mixing efficiency between these flows and the fuel may be sub optimal. This hypothesis needs to be experimentally verified by conducting detailed studies on the premixing tube to enhance our understanding and optimization of the mixing process.

A

PLANNING



Figure A.1: Planning for thesis project

B

P & ID SWIRL BURNER

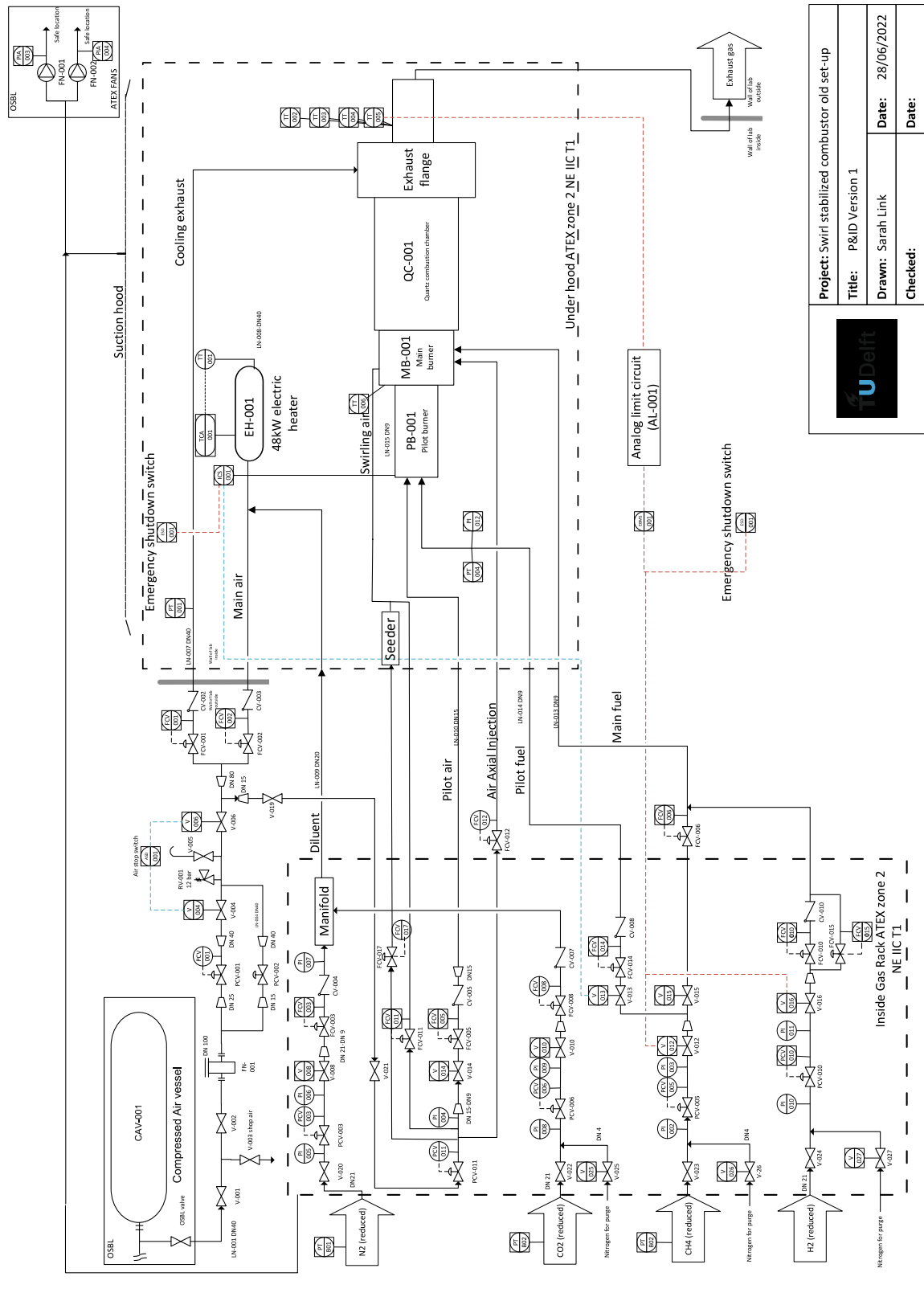
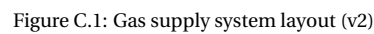


Figure B.1: P & ID swirl burner (V2)

Project: Swirl stabilized combustor old set-up	
Title: P&ID Version 1	Date: 28/06/2022
Drawn: Sarah Link	Checked:
Date:	

C

GAS SUPPLY SYSTEM LAYOUT



D

LABVIEW CODE

E

EMISSION CODE

```
1 %Plotting emission measurements
2 clear all
3 close all
4
5
6 % filename = 'O:\H2 enrichment studies in swirl stabilized flame\Data Logging\excel\
   Swirl Combustor\Excel\2023_03_17_10_43_362023.03.08_Swirl_Burner_39.xlsx';
7 filename='D:\Downloads\Delft (Vakken projecten resultaten)\Thesis\Code\Emissions\
   Swirl burner February 2024\2024_02_19_13_41_33P8_H25_AAI10_f60.tdms';
8
9 data = tdmsread(filename);
10
11 %%
12 POW=data{1}.("Main Burner Power (kW)");
13 XH2=data{1}.("Hydrogen Mix (%)");
14 AAI=data{1}.("AAI (%)");
15
16 tSteps=data{1}.Time;
17
18 CH4_main = data{7}.("Measured - CH4 Main ");
19
20 NO_measured=data{3}.NO;
21 CO2_measured = data{3}.CO2;
22 O2_measured = data{3}.O2;
23 CO_measured = data{3}.CO;
24 time=data{3}.("Time Stamp");
25
26
27
28 %%
29
30 %time windows for averaging
31 av_window = seconds(5);
32 av_window.Format = 'mm:ss.S';
33
34 time_setpoint_s = seconds(tSteps(:,1));
35 time_setpoint_s.Format = 'mm:ss.S';
36
37 time_s = seconds(time(:,1));
```

```

38 time_s.Format = 'mm:ss.S';
39
40
41 itSteps = [1]
42 avgNO=mean(NO_measured(itSteps:end));
43
44
45 figure ()
46
47 plot (time ,NO_measured, 'LineWidth',2)
48
49 grid on
50 hold on
51
52 plot (time(itSteps) ,NO_measured(itSteps) , 'r.' , 'MarkerSize' ,15)
53
54 for i=1:numel(itSteps)
55
56     if i~=numel(tSteps)
57         plot ([time(itSteps(i)) time(itSteps(i+1))],[avgNO(i) avgNO(i)] , 'LineWidth'
58             ,2)
59     elseif i==numel(tSteps)
60         plot ([time(itSteps(i)) time(end)],[avgNO(i) avgNO(i)] , 'LineWidth' ,2)
61
62     end
63
64     str=sprintf('POW=%.2f , AAI=%.2f , XH2=%.2f' ,POW(i) ,AAI(i) ,XH2(i));
65     text (time(itSteps(i)) ,avgNO(i)+0.05,str)
66
67 end
68
69 xlabel('time\ [s]' , 'FontSize' ,30, 'interpreter' , 'latex')
70 ylabel('NO' , 'FontSize' ,30, 'interpreter' , 'latex')
71
72 %% CH4 main
73
74 for i = 1:numel(time_setpoint_s)
75     time_stepxx = find(time_setpoint_s(i) == time_s);
76
77     time_window = seconds(30); % time window for steady state in seconds
78     time_window.Format = 'mm:ss.S';
79
80     if time_setpoint_s (i) ~= time_setpoint_s(end)
81         time_start = time_setpoint_s(i+1);
82     else
83         if any(main_burner2(time_stepxx:end) == 0)
84
85             %-1 here because it finds the first one which is zero, but we
86                 %want the last one which is 1
87             time_start = find(main_burner2(time_stepxx:end) == 0,1)-1;
88             %-1 here because we add +time_steps, but that one starts
89                 %counting at the last 0
90
91             time_start = time_start + time_stepxx -1;
92             time_start = seconds(time_start);

```

```

93         time_start.Format = 'mm:ss.S';
94     else
95         time_start = time_setpoint_s(end);
96     end
97 end
98
99 % initial values for NO to trigger while loop
100 CH4_measured_1 = 1;
101 CH4_measured_2 = 0;
102
103
104 %treshhold for steady state conditions (TO BE DISCUSSED)
105
106 while abs(CH4_measured_1 - CH4_measured_2)/CH4_measured_1 > 0.05
107
108     time_end = time_start-time_window;
109     [minValue_start,time_step_start] = min(abs(time_s - time_start));
110     [minValue_end,time_step_end] = min(abs(time_s - time_end));
111
112 %averaging window of 5s at beginning and end of the 30s
113
114     [minValue_time_5s1,time_5s1] = min(abs(time_end+av_window-time_s));
115     [minValue_time_5s2,time_5s2] = min(abs(time_start-av_window-time_s));
116
117
118     CH4_measured_1 = mean(CH4_main(time_step_end:time_5s1));
119     CH4_measured_2 = mean(CH4_main(time_5s2:time_step_start));
120
121     CH4_average(i) = mean(CH4_main(time_step_end:time_step_start));
122     CH4_stdv (i) = std(CH4_main(time_step_end:time_step_start));
123
124
125     time_window = time_window - av_window;
126
127     if time_window < seconds(15)
128         clear str
129         %disp(num2str(P_th), num2str(AAI_m), num2str(X_H2_m), 'Averaging window
130             too short for steady state, no steady state found')
131         disp('Averaging window too short' + str);
132     break
133     else
134     end
135 end
136
137 end
138
139 %%
140 figure ()
141
142 plot(time,CH4_main,'LineWidth',2)
143
144 grid on
145 hold on
146
147 plot(time(itSteps),CH4_main(itSteps),'r.','MarkerSize',15)

```



```

148
149 for i=1:numel(itSteps)
150
151     if i~=numel(tSteps)
152         plot([time(itSteps(i)) time(itSteps(i+1))],[CH4_average(i) CH4_average(i)],'
153             LineWidth',2)
154         %plot([time(itSteps(i)) time(itSteps(i+1))],[CH4_stdv(i) CH4_stdv(i)],'
155             LineWidth',2)
156         errorbar(time(itSteps(i+1)-150), CH4_average(i) ,CH4_stdv(i), 'LineWidth',1,
157             'Color', 'b')
158
159     elseif i==numel(tSteps)
160
161         plot([time(itSteps(i)) time(end)],[CH4_average(i) CH4_average(i)],'LineWidth
162             ',2)
163         errorbar(time(end)-150, CH4_average(i) ,CH4_stdv(i), 'LineWidth',1, 'Color',
164             'b')
165
166     end
167
168     str=sprintf('P=%0f,AAI=%0f,H2=%0f',POW(i),AAI(i),XH2(i));
169     text(time(itSteps(i)),CH4_average(i)+0.7,str, 'FontSize',8)
170
171     str2 = sprintf('RMS =%0.2f', CH4_stdv(i));
172     text(time(itSteps(i))+20,CH4_average(i)-0.7,str2, 'FontSize',8)
173
174 end
175
176 xlabel('time\ [s]', 'FontSize',30, 'interpreter', 'latex')
177 ylabel('CH4 [nlpm]', 'FontSize',30, 'interpreter', 'latex')
178
179
180
181 figure();
182 plot(time, T_exhaust_1);
183 hold on
184 plot(time, T_exhaust_2);
185 plot(time, T_exhaust_3);
186 plot(time, T_exhaust_4);
187 plot(time(itSteps),T_exhaust_4(itSteps),'r.', 'MarkerSize',15)
188
189 xlabel('time\ [s]', 'FontSize',30, 'interpreter', 'latex')
190 ylabel('Exhaust Temperature [K]', 'FontSize',30, 'interpreter', 'latex')

```

F

ACOUSTIC MEASUREMENTS

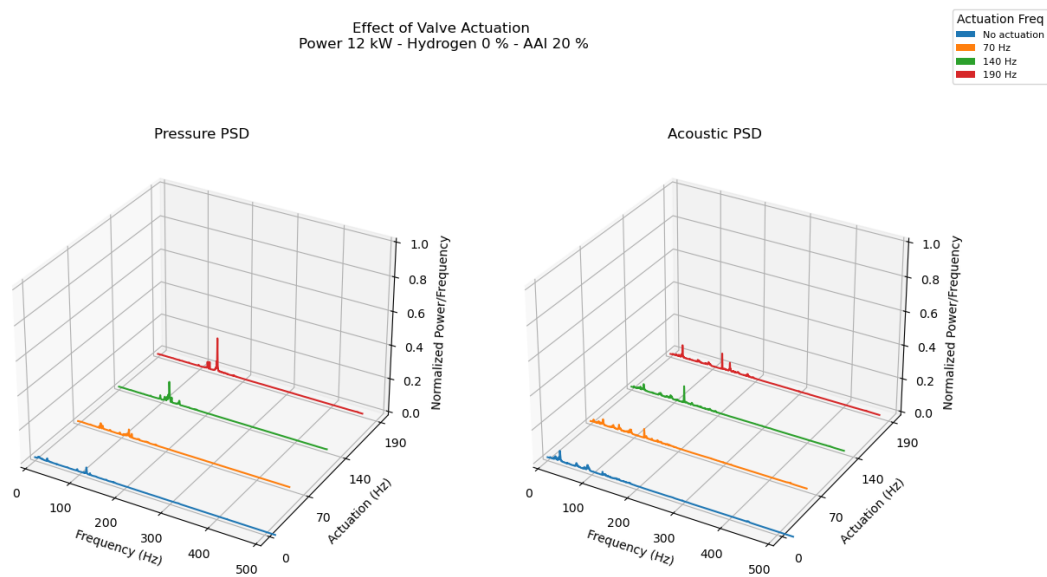


Figure F.1: Effect of different actuation frequencies on acoustics
(P:12 kW, H₂: 0 %, AAI:20%)

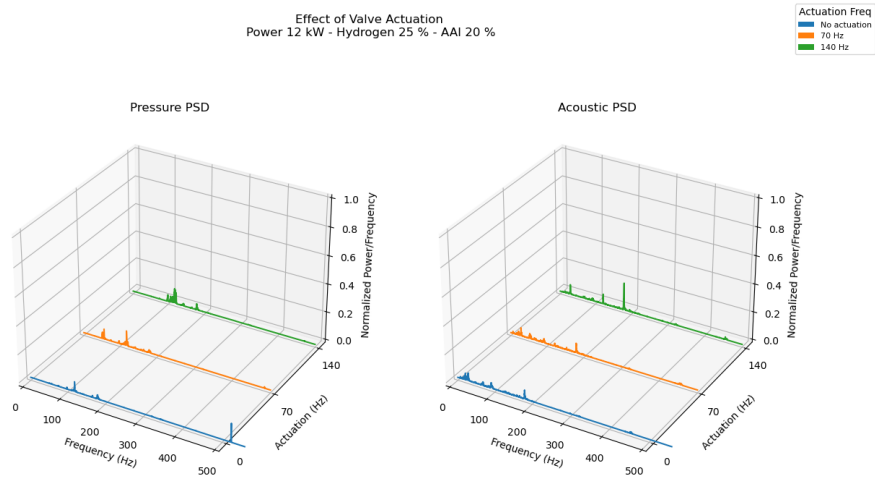


Figure E2: Effect of different actuation frequencies on acoustics
(P:12 kW, H2: 25 %, AAI:20%)

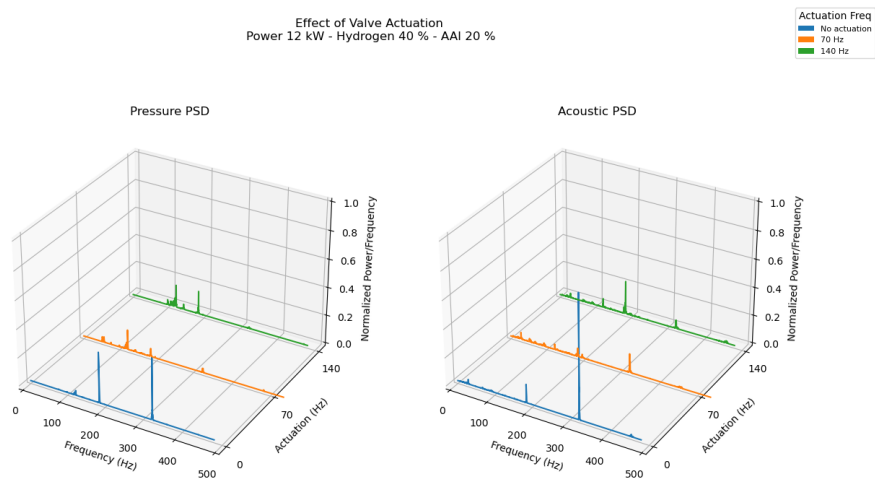


Figure E3: Effect of different actuation frequencies on acoustics
(P:12 kW, H2: 40 %, AAI:20%)

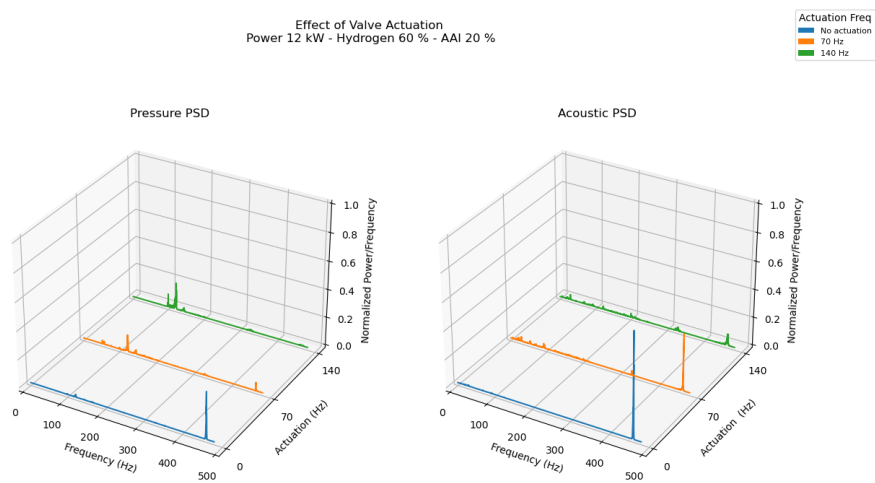


Figure E4: Effect of different actuation frequencies on acoustics
(P:12 kW, H2: 60 %, AAI:20%)

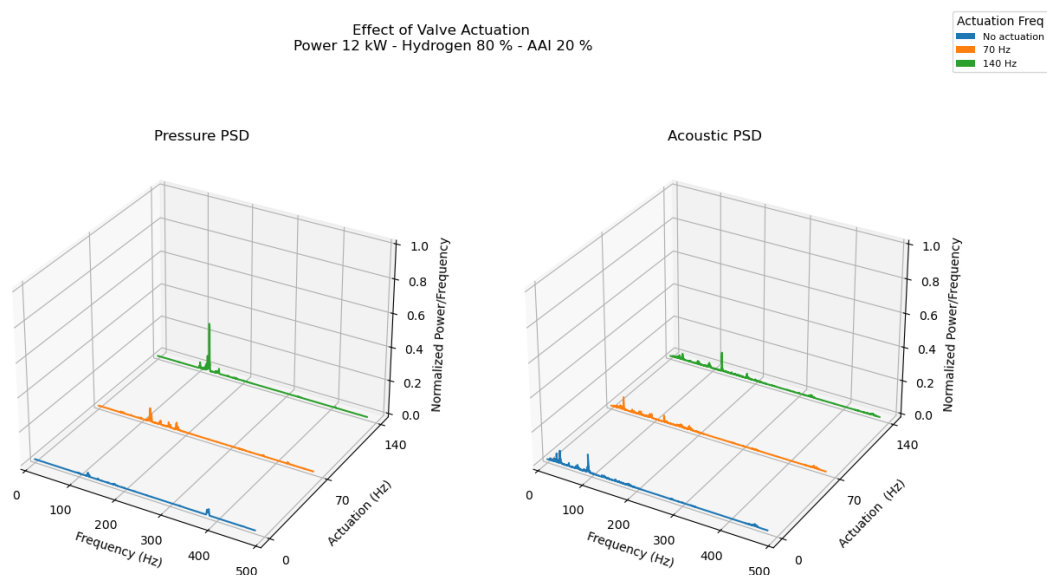


Figure F5: Effect of different actuation frequencies on acoustics
(P:12 kW, H₂: 80 %, AAI:20%)

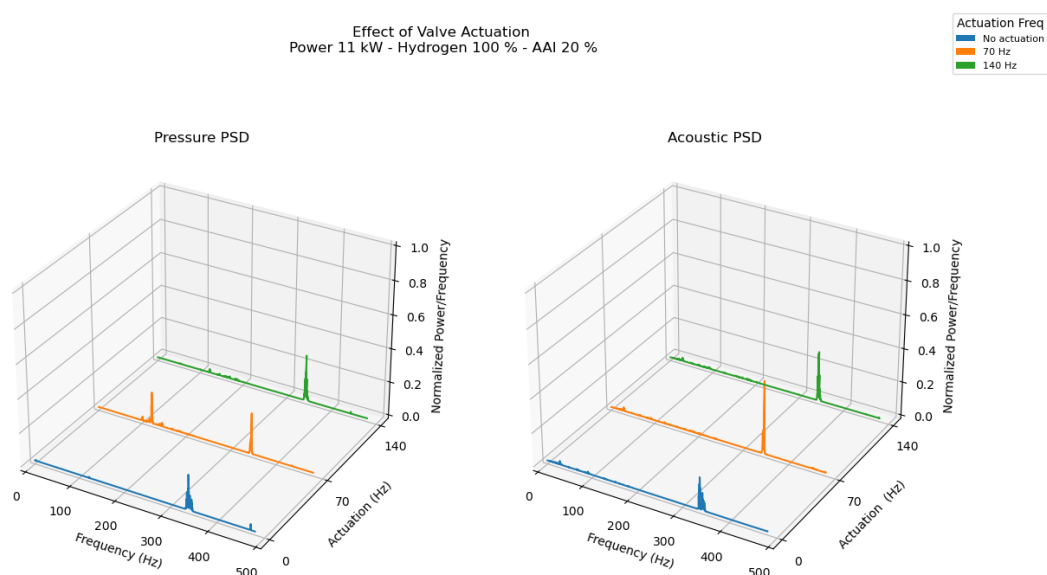


Figure F6: Effect of different actuation frequencies on acoustics
(P:11 kW, H₂: 100 %, AAI:20%)

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