

Investigation of Zero-Emission Hydrogen Oxyfuel Flames

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Zusammenfassung

Energiespeicherung gilt als eine der großen Aufgaben bei der Energiewende. Quellen wie Wind und Sonne sind zwar erneuerbar aber nicht konstant. Sie erfordern daher eine Zwischenspeicherung um Produktion und Verbrauch zu synchronisieren. Ein vielversprechendes Speichermedium ist sog. grüner Wasserstoff, der aus Wind- oder Solarstrom hergestellt wird. Für die Rückverstromung eignet sich neben der Brennstoffzelle die Verwendung von Verbrennungstechnologie. Zu diesem Zweck wurde in der vorliegenden Arbeit ein Laborbrenner entwickelt und untersucht, der Wasserstoff (H_2) und Sauerstoff (O_2) im stöchiometrischen Verhältnis zu Wasserdampf verbrennt, ohne dabei umweltschädliche Stoffe auszustoßen. Der Brenner ermöglicht somit die emissionsfreie Umwandlung von gespeichertem Wasserstoff in Strom und Wärme. Die Entwicklung des Brenners basiert auf der drall-stabilisierten Verbrennung, wie sie als optimiertes und erprobtes Konzept in modernen Gasturbinen üblich ist. Allerdings birgt die extrem hohe Reaktivität von H_2/O_2 -Gemischen sehr viel größere Herausforderungen als die Verbrennung von fossilen Brennstoffen in Luft. Dazu zählen vor allem die hohen Flammentemperaturen und Brenngeschwindigkeiten, sowie die notwendige Verbrennungseffizienz. Im Rahmen der Arbeit wurden umfangreiche Studien zum atmosphärischen Brennverhalten des entwickelten Brenners durchgeführt. Es wird gezeigt, dass Flammen- und Wandtemperatur der Brennkammer durch Verdünnung der Flamme mit Dampf stark reduziert werden können, was die thermische Belastung des Systems reduziert. Der Grad der Verdünnung Ω ist dabei ein entscheidender Parameter in dieser Arbeit, da er gemeinsam mit der Flammenleistung die Dampftemperatur am Auslass bestimmt. Diese ist für den Operationsbereich des Brenners in der realen Anwendung ausschlaggebend. Bei ausreichender Drallstärke der Strömung, ermöglicht der entwickelte Brenner eine stabile Drallflamme über einen sehr breiten Bereich von Ω . Die obere Grenze des Einsatzbereiches liegt etwa bei einer Verdünnung von $\Omega = 6.5$, was bedeutet, dass 6.5-mal mehr Wasserdampf zur Verdünnung dazu gegeben wird, als die Flamme selbst erzeugt. Flammen mit solch hohem Ω lassen sich optimal zur Über- und Zwischenerhitzung von Dampfströmen in einem Kreisprozess einsetzen, wo dem Dampfstrom primär Wärme und nicht Masse hinzugeführt werden soll. Die Arbeit zeigt, dass die obere Grenze dabei nicht durch das Erlöschen der Flamme gegeben ist, sondern durch ihre Verbrennungseffizienz, die bei zu hoher Verdünnung abfällt. Eine hohe Verbrennungseffizienz ist allerdings in einem H_2/O_2 -Brenner absolut notwendig, da der produzierte Dampf sonst H_2 und O_2 enthalten kann, was die restlichen Komponenten des Kraftwerkes beschädigen würde. Für die Evaluierung der Verbrennungseffizienz wird in der Arbeit, zum ersten Mal bei H_2/O_2 Flammen, ein zeitaufgelöstes System genutzt. Das System basiert auf der Bestimmung der O_2 Konzentration im Abgas mittels Absorptionsspektroskopie und die durchgeführten Messungen legen nahe, dass sich das System vor allem im gekoppelten Einsatz mit einer Lambdasonde bewährt. Mit diesem Messsystem ist es möglich die Verbrennungseffizienz live zu überwachen, was für die Verwendung in Kraftwerksprozessen sehr vorteilhaft ist. Der Brenner besitzt bei den atmosphärischen Tests in dieser Arbeit noch keine perfekte Verbrennungseffizienz. Eine Parameterstudie des Bren-

nerdesigns zeigt jedoch, dass die Verbrennungseffizienz durch erhöhten Drall und eine höhere Leistungsdichte gesteigert werden kann. Gleichzeitig weist der Brenner auch bei höherer Leistungsdichte keinerlei Beschädigungen oder Verschleißerscheinungen auf. Basierend auf den Erkenntnissen dieser Arbeit, wird die technische Umsetzbarkeit der dampfverdünnten H_2/O_2 -Verbrennung für Kraftwerksprozesse als sehr positiv eingeschätzt.

Abstract

Energy storage is regarded as one of the major tasks in the transition from fossil to renewable energy generation. Sources such as wind and sun are volatile and require intermediate storage to synchronize production and consumption. A promising storage medium is hydrogen (H_2), which is produced from wind or solar power. Besides the fuel cell, combustion is a suitable technology for re-generating electricity from H_2 . For this purpose, a laboratory burner was developed and investigated in the present thesis. The burner converts H_2 and oxygen (O_2) in the stoichiometric ratio to water vapor without generating environmentally harmful emissions. The burner thus enables the emission-free conversion of stored hydrogen into electricity and heat. Its development was based on swirl-stabilized combustion, which is an optimized and proven concept common in modern gas turbines. However, the combustion of H_2 and O_2 is much more challenging than burning fossil fuels due to its exceptionally high reactivity. The challenges include the high flame temperature and speed, and the required combustion efficiency. In the course of the work, extensive studies were carried out on the atmospheric combustion behavior of the developed burner. It is shown that flame and wall temperatures of the combustion chamber can be significantly lowered by diluting the flame with steam, which reduces the thermal load on the system walls. The degree of dilution Ω is a crucial parameter in this thesis because it determines (together with the flame power) the combustor outlet temperature, which is the decisive boundary condition in real applications. With sufficient swirl intensity of the flow, the developed burner allows a stable swirl flame over a wide range of Ω . The upper limit of this range is approximately at a dilution of $\Omega = 6.5$, which means that 6.5 times more steam is added as dilution than the flame produces itself. Flames with such a high Ω can be optimally used for super- and intermediate heating of steam flows in a cyclic process where heat rather than mass addition is required. The thesis shows that the upper limit of Ω is not given by the cold blow-off, but by the flame's combustion efficiency, which drops when the flame is diluted too strong. However, high combustion efficiency is necessary for an H_2/O_2 burner, because otherwise, the produced steam could contain H_2 and O_2 , which would damage the remaining components of the power plant. In the thesis, a system is deployed that evaluates the combustion efficiency time-resolved, which was not done before for H_2/O_2 flames. The system is based on the determination of the O_2 concentration in the flue gas by absorption spectroscopy, and the measurements performed suggest that the system is particularly suitable in combination with a lambda sensor. Such a measurement system enables live monitoring of the combustion efficiency, which is very advantageous for the power plant operation. During the atmospheric tests conducted in this thesis, the burner did not present a perfect combustion efficiency. However, a parameter study of the burner design indicates that enhanced swirl intensity and a higher power density can increase the combustion efficiency. At the same time, the burner shows no signs of damage or fatigue even at higher power density. Based on the findings of this thesis, the technical feasibility of steam-diluted H_2/O_2 combustion for power plant processes is assessed as very positive.

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Nomenclature

Latin symbols

I to XIV	burner configuration numbers
A	area (for A_{H_2} , A_{O_2} , A_{ISD} , A_{OSD} see Tab. 2.8)
A_{in}	inlet area of the combustion chamber
b	blocking ratio of the swirl generator ducts
cp	specific heat capacity at constant pressure
C_{i}	concentration of H_2 or O_2 in the water tunnel
C_{Mix}	normalized time-averaged concentration of combined H_2 and O_2 in the water tunnel
$C_{\text{Mix,RMS}}$	relative RMS concentration of superposed H_2 and O_2 in the water tunnel
d_{duct}	inner (i), outer (o) and hydraulic (h) diameter of the cooling duct
D	diameter of the combustion chamber
D_{H_2}	diameter of the H_2 injector
$D_{\text{H}_2\text{O}}$	hydraulic diameter of the steam ducts
D_{O_2}	diameter of the O_2 injectors
D_{IRZ}	diameter of the inner recirculation zone
G_{ax}	axial momentum
G_{t}	tangential momentum
h	specific enthalpy
$\dot{H}$	enthalpy flux
I_{cp}	deconvoluted OH^* intensity in the center plane
I_{OH^*}	line-of-sight integrated OH^* intensity
L_{cc}	combustion chamber length
L_{duct}	cooling duct length

L_f	flame length
L_{IRZ}	length of the inner recirculation zone
L_{op}	optical path length of the TDLAS system
L_p	penetration depth of the H_2 jet into the IRZ
$\dot{m}_{\text{air}}$	mass flow of air
$\dot{m}_{\text{ct}}$	mass flow in the combustion test rig for H_2 , O_2 or steam
Nu	nusselt number at the inner combustion chamber wall
Nu_{duct}	nusselt number at the outer combustion chamber wall
p	static pressure
$P(u < 0)$	back flow probability
P_{th}	thermal power
P_{w}	power output of wind turbines
P_{wt}	thermal power equivalent in the water tunnel
q	dynamic pressure
$\dot{Q}$	heat flux
R_{i}	inner radius of the steam ducts
R_{o}	outer radius of the steam ducts
Re_{ct}	Reynolds numbers in the combustion test rig
Re_{wt}	Reynolds numbers in the water tunnel
s	stoichiometric O_2/H_2 mass ratio
s_{L}	laminar flame speed
s_{T}	turbulent flame speed
s_{w}	wall thickness of the combustion chamber
S	swirl number of the flow at the combustion chamber inlet
T_{ad}	adiabatic flame temperature
T_{ex}	exhaust temperature
T_{iw}	inner combustion chamber wall temperature
T_{LS}	lambda sensor temperature
T_{ow}	outer combustion chamber wall temperature
T_{plenum}	plenum temperature

u	axial velocity
u_0	bulk velocity in the combustion chamber
$\overline{u_i}$	time-averaged velocity component
u'_i	velocity fluctuation component
u_{RMS}	turbulence intensity in axial direction
U_{LS}	lambda sensor voltage
U_s	spatial unmixedness factor
U_t	temporal unmixedness factor
$\dot{V}_{\text{wt}}$	volume flows in the water tunnel
v_{O_2}	O ₂ volume fraction
w	mass fraction
x	axial direction
x_{f}	axial flame position
x_{us}	axial position of the inner recirculation zone
y	radial direction

Greek symbols

$\alpha_{\text{air}}, \alpha_{\text{H}_2\text{O}}$	heat transfer coefficients of the cooling air and the steam
α_{f}	flame angle
Δp	pressure loss
η_c	combustion efficiency
λ	thermal conductivity
ν_{ct}	kinematic viscosity in the combustion test rig for H ₂ , O ₂ or steam
ν_{wt}	kinematic viscosity of the water in the water tunnel
Π	oxygen enrichment ratio
ρ_{ct}	density in the combustion test rig for H ₂ , O ₂ or steam
ρ_{wt}	density of the water in the water tunnel
σ	uncertainty, standard deviation
ϕ	equivalence ratio
χ	swirler splitting ratio

Ω	steam dilution ratio for H ₂ /O ₂ combustion
Ω^*	steam dilution ratio for H ₂ /O ₂ /air combustion
Ω^+	steam and N ₂ dilution ratio for H ₂ /O ₂ /air combustion

Other symbols

% _m	mass fraction
% _v	volume fraction

Acronyms

AEL	alkaline electrolysis
BC	boundary condition
CAES	compressed air energy storage
CC	combustion chamber
CFD	computational fluid dynamics
CH*	methyldyne radical
CH ₄	methan
CO	carbon monoxide
CO ₂	carbon dioxide
Conf.	burner configuration
CTR A	combustion test rig A
CTR B	combustion test rig B
DLR	Deutsches Zentrum für Luft- und Raumfahrt (German Aerospace Center)
ENTSO-E	European Network of Transmission System Operators for Electricity
EU	European Union
FES	flywheel energy storage
FC	fuel cell
GC	gas chromatograph
H ₂	hydrogen
H ₂ /O ₂ -SG	H ₂ /O ₂ steam generator
HHV	higher heating value
HRSG	heat recovery steam generator

HP	HP burner: fuel-flexible burner with axial air injection
HP-T	high-pressure turbine
HT-T	high-temperature turbine
IEA	International Energy Agency
IP-T	intermediate-pressure turbine
IRZ	inner recirculation zone
IS	inner swirler
ISD	inner steam duct
LCOE	levelized cost of electricity
LHV	lower heating value
LP-T	low-pressure turbine
LS	lambda sensor
MB	movable block burner: fuel-flexible burner with centerbody
N ₂	nitrogen
NO _x	nitrogen oxides
NOW	NOW GmbH Nationale Organisation Wasserstoff- und Brennstoffzellentechnologie
NPV	net present value
O ₂	oxygen
OH*	hydroxyl radical
OS	outer swirler
ORZ	outer recirculation zone
OSD	outer steam duct
PCR	primary control reserve
PDF	probability density function
PEM	polymer electrolyte membrane electrolysis
PHS	pumped hydro storage
PID	proportional integral derivative controller
PIV	particle image velocimetry
PLIF	planar laser-induced fluorescence

PV	photovoltaic
RANS	Reynolds-averaged Navier-Stokes
RMS	root mean square
SCR	secondary control reserve
SL	shear layer
SOEC	solid oxide electrolyzer cell
TCR	tertiary control reserve
TDLAS	tunable diode laser absorption spectroscopy
TSO	transmission system operator
UCTE	Union for the Co-ordination of Transmission of Electricity
UHCs	unburned hydrocarbons
UN	United Nations
VB	vortex breakdown
WMS	wavelength modulation spectroscopy
WT	water tunnel

CHAPTER 1

Introduction

The vast majority of researchers in the world declare the fight against climate change as the most significant challenge of our generation. In the Paris agreement in 2015 [190], the United Nations (UN) defined the goal of limiting global warming to two degrees. Today, most countries commit to the agreement and the challenging measures necessary. To meet these commitments, the emission of greenhouse gases like carbon dioxide (CO_2) must be reduced dramatically. A worldwide technological paradigm shift from fossil fuel utilization towards renewable energy generation seems to be without any alternative.

In 2020, this transition in the energy (and particularly in the electricity) sector is no longer a future perspective but ongoing in many countries. The first chapter of the thesis provides a short overview of developments in the energy sector. However, the necessary technology to fully complete the transition is far from ready for the market yet, and in many fields, research still seeks for solutions. One of these fields with particular demand for innovations is energy storage. Section 1.3 reviews the most common techniques for storing electrical energy.

In this context of energy storage, the thesis explores the emission-free combustion of the storage medium hydrogen (H_2). This technology facilitates the environmentally-friendly back-conversion of stored energy. Building upon previous research actions on similar technologies (see Sec. 1.4), a new combustor type for hydrogen combustion was developed and intensively studied on an application-oriented perspective. The thesis reports on the development and evaluation process in several steps from initial flow and mixing visualization to the analysis of the combustion characteristics. The last section of the introduction gives a more detailed overview of the thesis's purpose and structure.

1.1 Renewable Energy Generation

Even though most governments are aware of climate change and the need to reduce greenhouse gas emissions, the total worldwide energy consumption had increased by 1.7 % each year from approximately 300 EJ in 2005 to 355 EJ in 2015 [149]. A large share of about 20 % of this energy is consumed for electricity generation – today and in the future. Figure 1.1 (left) shows a forecast of the electricity consumption in 2030 in Germany and France. Based on a reference scenario, the consumption will stay approximately constant, and even with a low consumption scenario, no significant reduction can be expected [4]. Therefore, the necessary curtailment of greenhouse gas emissions must be reached mostly on the generation side.

Consequently, in 2017 nearly 70 % of the global investments in new power capacity for electricity generation were made in renewable technologies [149]. Figure 1.1 (right) shows

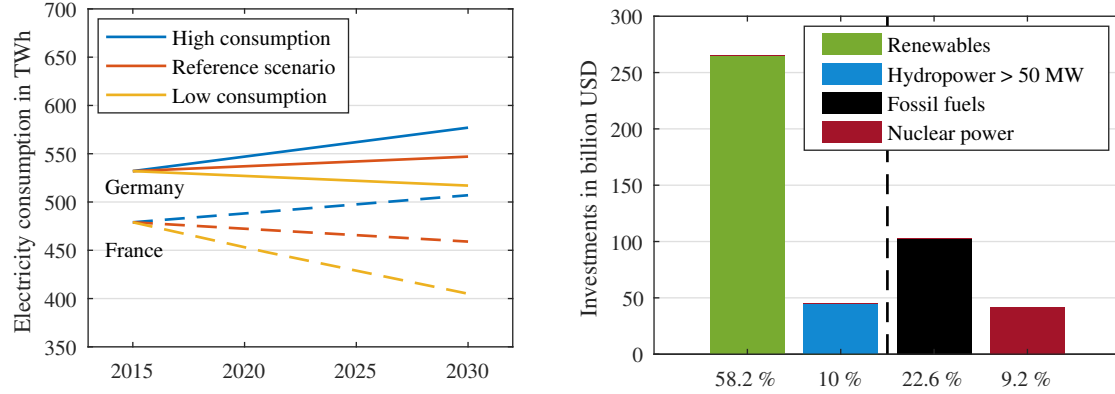


Figure 1.1: Left: Forecast of the electricity consumption in Germany (solid, net consumption) according to the German Grid Development Plan 2017 and France(dashed, gross consumption) according to the scenarios of the 2016 Bilan prévisionnel (extrapolated from 2021) ([4] p.41). Right: Global investments in new power capacity by type in 2017. The renewables are excluding biofuels and large (> 50 MW) hydropower projects ([149] p.146).

that approximately USD 265 billion were invested in electricity generation from wind, solar, and small hydropower plants. Another USD 45 billion were raised for large-scale hydropower (> 50 MW) projects [149].

The total amount of renewable electricity generation in the world has more than doubled in the last decade (2007-2017), as shown in Fig. 1.2. In 2017 it reached nearly 2.2 GW. However, more than half of the renewable electricity is generated from hydropower (Tab. 1.1), which will soon be limited in further expansion due to geological constraints. Indeed, it has increased much less than solar and wind power, which show the most substantial growth rates in Fig. 1.2. Even though their total share in worldwide electricity production is still low (Tab. 1.1), wind and solar energy are emerging contributors and will soon have a significant share in some countries. They are volatile energy sources, which can not be predicted sufficiently. When integrated into the electricity grid, they raise specific demands for the load and generation balancing, which is introduced in section 1.2.

Table 1.1: Estimated renewable energy share of global electricity generation 2017

Technology	Share
Non-renewable electricity	73.5 %
Hydropower	16.4 %
Wind power	2.2 %
Solar PV	1.9 %
Ocean, CSP and geothermal power	0.4 %

Solar Photovoltaic Energy

The global solar photovoltaic (PV) capacity is exponentially increasing, as displayed in Fig. 1.3. In 2017 it reached a total amount of 402 GW [149]. China (32.6 %) had the largest capacity for solar PV due to a massive expansion in 2017 when it raised 53.3 % of the global addition. It is followed by the United States, Japan, and Germany, with market shares of 10-12 %. According to the global market outlook by SolarPower Europe [164], the annual addition of solar PV will keep increasing from 99.1 GW in 2017 to 157.8 GW in 2022, which

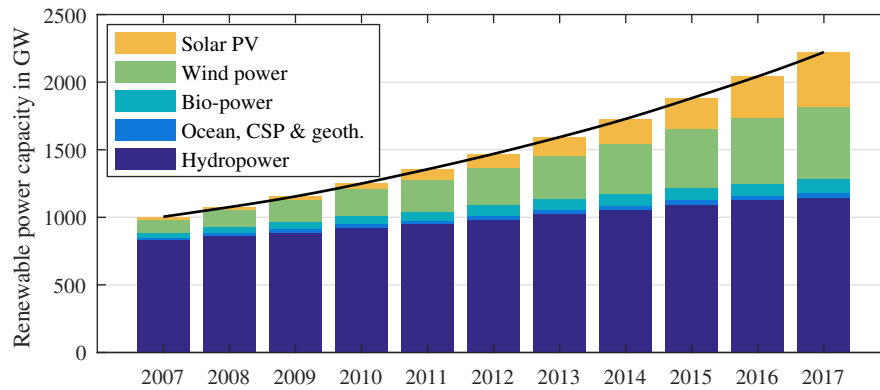


Figure 1.2: Global renewable power capacity 2007-2017. [149] p.41

means that the global capacity is doubled within four years. Regional circumstances like sun intensity and available space strongly affect the expansion of PV. The regions of the highest potential might not match the high-demand areas, which raises the challenge of storing and transporting renewable energy (Sec. 1.3).

In Germany, the expansion of renewable electricity generation is regulated and supported by the National Renewable Energies Action Plan (NREAP). Thereby, the development of solar PV capacity in Germany reveals an interesting trend. Figure 1.4 shows that the annual additions to the capacity increased strongly until 2012 when the addition was about 8.2 GW, and the total capacity reached 34 GW [140]. In 2012 an amendment of the governmental PV expansion support came into force, which defined a deployment corridor of 2.5-3.5 GW annual addition [172]. This example shows how volatile the development of renewables is and that it depends much on political decisions.

Wind energy

Figure 1.3 shows that in contrast to PV, the wind power capacity is rising linearly instead of exponentially with annual additions of 52-64 GW since 2014 [149]. As before, China has by far the highest amount of installed wind capacity with a market share of 34.9 %, followed by the United States (16.5 %), Germany (10.4 %), and India (6.1 %). Thus, more than two-thirds of the global wind power capacity is installed in these four countries. However,

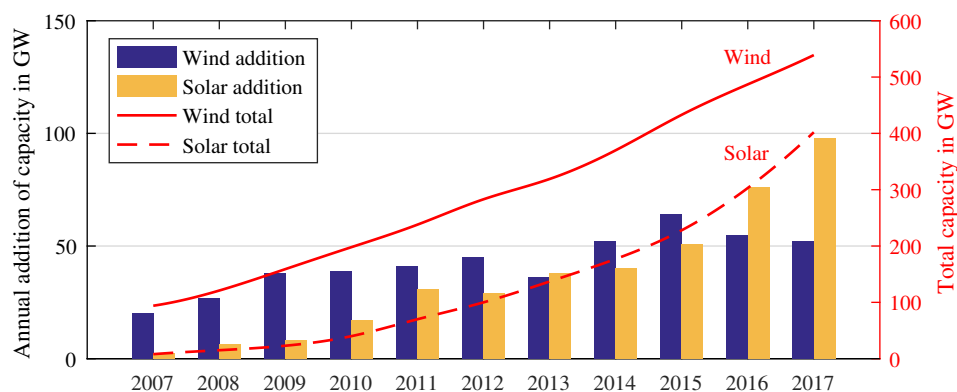


Figure 1.3: Global wind turbine (-) and solar capacity (- -) [149]. The bars show the capacity of the previous year and the annual addition.

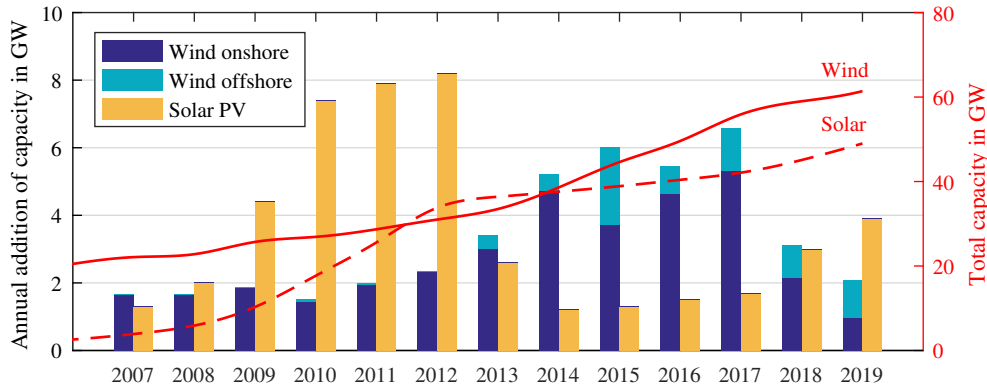


Figure 1.4: Annual capacity addition (bars) and cumulative capacity (line) of wind (solid) and solar PV (dashed) in Germany ([48, 49, 140]).

this concentration will probably soon dilute to less densely populated regions as there is a lack of suitable space and, at least in Germany, increasing resistance in public acceptance of wind energy expansion. Due to new regulations, the annual additions of wind power generation in Germany dropped 2018 and 2019 to 3 and 2 GW (Fig. 1.4). In the future, renewable energy might not be entirely generated in Germany but mainly be imported from abroad.

Since neither solar PV modules nor wind turbines operate in continuous base load, the actual generated gross power is not directly related to the installed capacity. For instance, wind turbines have a higher average load level compared with solar PV. Thus, the net power generation of the wind turbines in 2017 was nearly three times the one of solar PV. In Fig. 1.5 the gross power generation of all renewable energy sources in Germany is predicted until 2030 [2]. Its total amount will double between 2016 and 2030, and the volatile sources solar and wind will have by far the largest share.

Fluctuations of Solar and Wind Energy

Based on the mentioned development, generation fluctuations caused by weather and season changes are likely to play an increasing role in the management of future electricity grids. The seasonal generation fluctuations predicted for 2030 are quite balanced in the European Union (EU), as shown in Fig. 1.6. Moreover, they can be forecasted well, which allows an

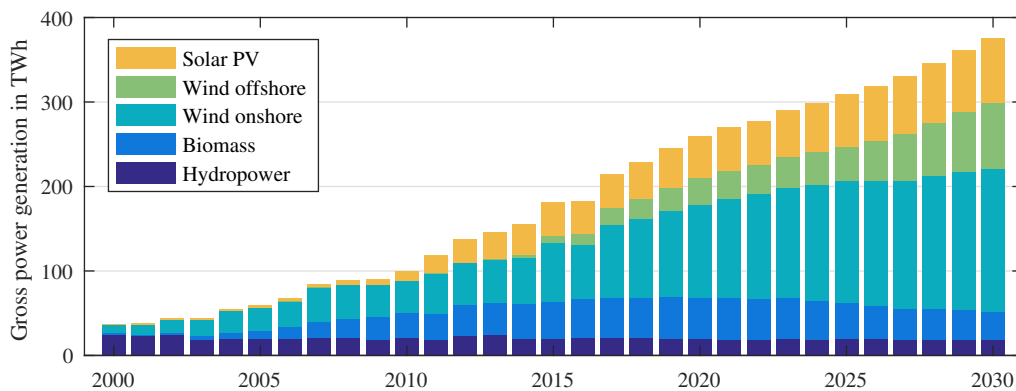


Figure 1.5: Prediction of the gross power generation by renewable sources in Germany from 2017. [2] p.53

accurate adaptation of the total generation on a monthly or weekly basis.

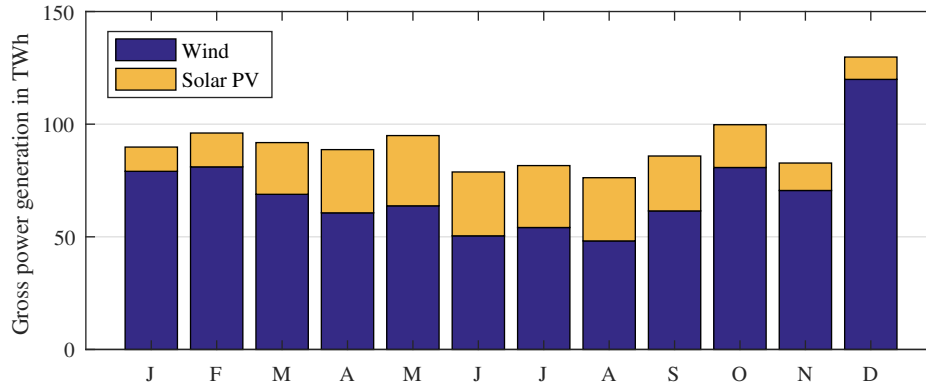


Figure 1.6: Prediction of the seasonal power generation by wind and solar PV in the EU 2030. Meteorological data from 2011. [60] p.7

Fluctuations due to weather changes are less predictable and happen suddenly. Thus they are influencing the daily energy generation and need appropriate handling. Figure 1.7 shows these short-term fluctuations for two exemplary weeks. On the right-hand side, the energy generation from renewable and conventional sources is depicted for calendar week 42, the week with the lowest share of renewable generation in 2018. During this particular week, the most fluctuation in production stemmed from solar energy peaking at noon and vanishing at night. This extra generation during the daytime coincides well with the maximum energy demand. The share of wind energy in this week is meager.

On the contrary, the left-hand side shows the same plot for calendar week 11 in 2019, the week with the highest renewable share in Germany (as of March 2019). Wind turbines contributed more than half of the energy generated in this week. Its share ranged from 25 GW to 55 GW and was not governed by the daytime but by weather conditions. Thus it did not coincide with the consumption, plotted as the red line in Fig. 1.7. In calendar week 11, there was a clear overproduction due to the high amount of renewable power generation. Hence, the amount of exported energy to the neighboring countries is twice as high as in week 42. Even more significant was the impact on the electricity price, which was enlarged by a factor of 2.6, as shown in Tab. 1.2. Such fluctuations of the electricity price offer the possibility of power arbitrage if suited energy storage is available.

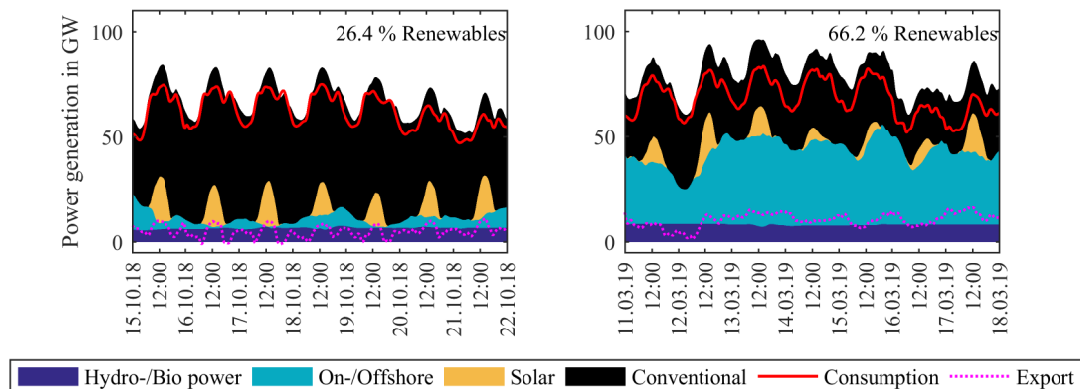


Figure 1.7: Power generation in Germany in calendar week 42 in 2018 (left) and calendar week 11 in 2019 (right). [3, 59]

Table 1.2: Power generation, renewable share, export saldo and electricity price in Germany for to specific calender weeks. [3, 59]

CW, Year	Generation	Consumption	Renewables	Export	Electricity price
42, 2018	11.3 TWh	10.5 TWh	26.4 %	0.85 TWh	64.27 €/MWh
11, 2019	13.3 TWh	11.5 TWh	66.2 %	1.74 TWh	24.40 €/MWh

1.2 Grid Balancing

Interconnected grids provide electricity for industry and households. The stability of the electricity supply in these so-called synchronous areas must be observed and controlled continuously. Exemplarily, the thesis refers to the Union for the Co-ordination of Transmission of Electricity (UCTE) grid, which is operated since 2009 by European Network of Transmission System Operators for Electricity (ENTSO-E). Their task is to maintain the stability of the UCTE grid using frequency control [189]. Therefore, the nominal net frequency of $f_0 = 50$ Hz is kept as constant as possible, which is done by four control measures:

- primary control reserve (PCR) [189] or frequency containment process [58],
- secondary control reserve (SCR) [189] or frequency restoration process [58],
- tertiary control reserve (TCR) [189] or reserve replacement process [58],
- and time control.

Primary Control Reserve

The PCR aims to balance the generation and consumption in the grid and is a joint action of all transmission system operators (TSOs). It stabilizes the system frequency f_s after a frequency deviation $\Delta f = f_0 - f_s$. However, it is not restoring the frequency to its nominal value f_0 . At $\Delta f = \pm 20$ Hz the PCR is activated first, reaching full activation at 200 Hz. Thereby, all available PCR in the UCTE grid should have a combined capacity of 3000 MW capacity to capture even incidences like the outage of a large power generation unit or a sudden drop in consumption. In any case, half of the total PCR must be deployed within 30 s and the rest within 60 s. It needs to deliver the additional power until SCR steps in with a minimum duration capability of 15 minutes.

In power plants using turbomachinery, the task of primary control is to keep the turbine speed constant. A commonly used method in coal and nuclear power plants is live steam throttling. Here, the steam flow from the boiler to the high-pressure turbine (HP-T) is reduced by a valve generating a steam reserve in the boiler. If PCR is required, the valve opens and increases the load of the turbines. Hence, when PCR is not activated, i.e., in baseload operation, the power plant operates not at its maximum capability, which increases capital costs and the specific fuel consumption [20, 62]. Hence, the plant's ecological footprint degrades. These disadvantages require the implementation of environmentally friendly and ultra-flexible alternatives that provide PCR in a stand-by manner.

Secondary and Tertiary Control Reserve

The SCR must be available for activation independently of the PCR. It controls the adjustable power generation and is active in a time frame of seconds up to 15 minutes after an incident [189]. The TSOs operate the SCR on individual blocks of the synchronous

area to maintain the power interchange of the blocks at the scheduled value and restore the nominal frequency f_0 in case of a deviation. This is achieved by an automatic PI (proportional-integral) controller, which operates online and in closed-loop. In the case of insufficient SCR, additional TCR is activated manually. It changes the generation or load in a block and must be able to cover the largest possible loss of generation. It can also free the SCR for long-time operation in a balanced system mode.

Market situation

Hirth and Ziegenhagen [84] gave an overview of the market development of control reserves in the UCTE and Germany. The control reserves are traded in auctions on a weekly (PCR and SCR) or daily (TCR) basis. In 2014, the required balancing capacity payments in Germany were approximately EUR 400 million, which is less than 0.8 €/MWh of consumed electricity or 0.25 % of the household electricity price. Interestingly, the payments (−50 %) and the capacities (−15 % from 5.3 GW in 2008) have dropped significantly between 2008 and 2014, even though the generation of renewables has almost tripled in this period. Nevertheless, the German energy monitoring report 2019 [30] showed that continuously more PCR was offered in the last decade, while the amount of SCR and TCR decreased.

Even though the expansion of renewable energy generation did not inevitably increase the capacity of control reserves, their integration remains an urgent challenge in Germany. This issue is emphasized by the significantly growing amount of curtailed renewable energy in Fig. 1.8. Curtailment means that renewable power generation units have to stop operation because the grid cannot handle the amount of energy being fed-in. In Germany, the amount of curtailed energy reached 5.4 GWh in 2018 and affected mainly onshore (72 %) and offshore (25 %) wind power plants [30]. Even if the units are not running during curtailment, the unit operators must be paid, which caused costs of EUR 719 million to the TSOs in 2018. For comparison, only EUR 123 million have been spent on PCR, SCR, and TCR in the same period. The curtailed energy equals nearly 3 % of the total renewable generation. Energy, which could be harvested without any emissions, is currently wasted and reveals a lack of electricity storage capabilities. In 2014, the International Energy Agency (IEA) estimated the necessary storage capacity to approximately 310 GW in the United States, Europe, China, and India to reach the CO₂ reduction targets [87].

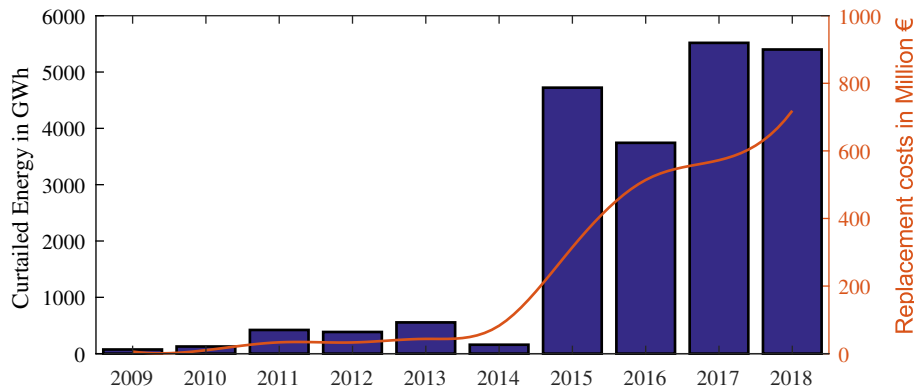


Figure 1.8: Curtailed renewable energy generation in Germany. [30]

1.3 Energy Storage Technologies

Energy storage is a key factor in modern electricity systems. There are plenty of review papers on storage technology and applications, especially in connection with the integration of renewable energy sources [5, 50, 79, 100, 121, 180, 204]. Large-scale energy storage offers a wide range of advantages, as summarized by Kousksou et al. [100].

- The excess generation by volatile sources (e.g., wind and solar power) can be stored,
- surplus electricity of low demand periods (e.g., overnight) can be stored and released at high demand,
- short-time demand peaks can be accommodated,
- ancillary services can be offered, reducing the necessary control reserve capabilities of power plants,
- and if more balancing is provided due to storage, less load adaptation can reduce the inefficient part-time operation of baseload power plants.

There are a lot of different technologies for energy storage with a variety of characteristics. However, each of them is a source of significant losses. An economical and ecological optimum must be found between flexible power generation and the usage of energy storage. The next paragraphs give an overview of some common techniques for electrical storage.

Pumped Hydro Storage

Pumped hydro storage (PHS) systems use the potential difference of an upper and a lower water reservoir. The water is pumped into the upper one while charging. If the electricity demand is high, the water flows down to a turbine, recovering the energy. The round trip efficiency of this process is usually around 65-75 % [100]. The duty cycle depends on the system and its purpose and can vary from daily to multiple times per hour. A notable advantage of PHS is the long lifetime of 30-50 years. On the other hand, it is associated with high capital costs as well as long planning and construction times. Finally, the erection of PHS systems depends strongly on the geographical suitability and environmental aspects.

Compressed Air Energy Storage

Compressed air energy storage (CAES) systems utilize an interrupted gas turbine cycle as energy storage. A motor consumes electrical energy to run a compressor. The compressed air is stored in natural aquifers, salt caverns, or reservoirs in rock formations, typically underground. Later, combustion heats up the air, which then drives a turbine. This process yields high storage efficiencies up to 70 %, but the combustion requires an additional energy source like natural gas, which makes it pollutive. Compared to PHS, the lifetime is almost the same (40 years) but with significantly lower capital costs and less impact on the surface environment [100].

Flywheel Energy Storage

A measure of short-term storage can be a flywheel energy storage (FES) with magnetic bearings in a vacuum chamber. It stores the electric energy in the form of the kinetic energy of a spinning wheel. Due to the low friction, the round trip efficiency is extremely high at

90-95 %. However, it is not suitable for long-term storage due to a high self-discharging rate. Moreover, the capital costs are high, and the energy density is low.

Battery Energy Storage

Batteries store electricity on an electrochemical basis. In general, they profit from short construction times and a high degree of modularity since they are built in relatively small units. On the contrary, this fact reduces the scaling effect in terms of costs. They also often suffer from small energy densities and power capabilities, high maintenance costs, and a short lifetime of only a few thousand cycles. There is a large variety of different types of batteries, but naturally only rechargeable ones are suited for the integration in electricity grids. A choice of suitable battery types and their efficiency is given in Tab. 1.3.

Table 1.3: Battery types and properties. [50, 100]

Battery type	Cost \$/kWh	Efficiency %	Characteristics
Lead-acid	300-600	70-90	low specific energy and power
Sodium-sulfur	350	85	high energy density, high life cycle
Lithium-based	900-1300	78-88	used for portable devices and cars
Flow batteries	> 150	75-85	very high life cycle (> 10000), scalable energy capacity (0.5-100 MW)

Hydrogen based energy storage

All the storage types, mentioned above, have a common disadvantage compared to fossil fuels. The stored energy is either not long-time storable (batteries, FES) or not transportable but has to be charged and discharged at the same location (PHS, CAES, FES). Due to a lack of natural resources, Germany imported 70 % of its primary energy carriers in 2019 [33]. Even with a 100 % renewable energy generation, it will be unlikely to generate all the required energy locally in Germany. The reasons are a lack of space, environmental boundary conditions, and the public acceptance. Renewable energy will be imported from less dense populated countries with a high solar or wind intensity. Hence, there is a strong need for large-scale, transportable energy storage. Viable alternatives are hydrogen (H_2) or synthetic fuels. H_2 is the cleanest possible fuel and has a very high specific energy density.

The first step of an H_2 energy storage is the gas production. An electrolyzer produces H_2 directly from electricity by the electrolytic dissociation of water. If renewably generated electricity is used, the process can be regarded as emission-free. Götz et al. [75] gave a technological and economic review of electrolyzers for renewable power-to-gas applications. There are three different types of electrolyzers: polymer electrolyte membrane electrolysis (PEM), alkaline electrolysis (AEL), and solid oxide electrolyzer cell (SOEC). Their properties are presented in Tab. 1.4. It has to be considered that the outlet pressure is expected to increase significantly in the next years [169]. A list of commercial electrolysis systems is given later in Tab. 2.6.

The second step, the H_2 storage, is most likely done in gaseous or liquid form in pressurized tanks or underground caverns. Further storage approaches by absorption in metal-hybrids or on carbon nano-fibers are under investigation [100]. During storage, pipelines and ships can easily transport H_2 even for long distances. This advantage could make H_2 and H_2 -based fuels one of the most important future energy carriers.

The last step of the process is the back-conversion into electricity, which is discussed in the next section.

Table 1.4: Estimated properties (NOW survey [169]) of different electrolyzer types : polymer electrolyte membrane electrolysis (PEM), alkaline electrolysis (AEL), and solid oxide electrolyzer cell (SOEC).

Fuel cell type	Capital cost €/kW	Energy consumption kWh/Nm ³	Outlet pressure bar	Life time h
PEM	1470	4.9	30	44000
AEL	920	4.6	18	57000
SOEC	2300	3.8	5	20000

1.4 Electricity Generation from Hydrogen

There are two major ways to generate electric power out of H₂: fuel cells (FCs) and combustion with either air or oxygen (O₂) as the oxidizer. Besides these two, H₂ can also act as a source for the production of synthetic fuels, which, for instance, could be fed to the natural gas grid and consumed by all conventional gas combustion systems without any technical adaptation [72]. However, while the utilization is straightforward, the synthetic gas production is complicated and connected to high installation and operation costs. Moreover, it requires CO₂ that must be captured from combustion processes or extracted from the atmosphere. Hence the overall cycle efficiency is very low. This technique is only a niche option for large-scale electricity storage [87, 174] and might be more suited in the transportation sector, eg., for aviation.

1.4.1 Fuel Cell

FCs are electrochemical cells that provide electricity from different hydrocarbons or H₂. Therefore, they reverse the electrolysis process and use either pure or atmospheric O₂ as the oxidizer. Sharaf and Orhan [165] provided a comprehensive review of different FC technologies concentrating on H₂ as fuel. They also provided a comparison of FCs to other electricity-generating technologies, for instance heat engines. The FC's advantage is that the chemical energy is directly converted into electricity. Hence, they provide relatively high efficiency, as shown in Tab. 1.5 for different types of FCs.

Table 1.5: Comparison of fuel cell types [195]: PEMFC (Polymer Electrolyte Membrane Fuel Cell), PAFC (Phosphoric Acid Fuel Cell), MCFC (Molten Carbonate Fuel Cell), SOFC (Solid Oxide Fuel Cell).

Fuel cell type	Typical size in kW	Temperature in °C	Efficiency in %
PEMFC	1-100	80	39-52
PAFC	100-400	300	38-22
MCFC	300-3000	650	40-55
SOFC	1-2000	900	45-60

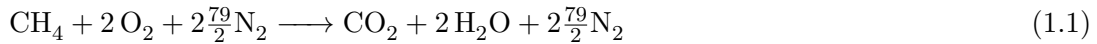
In general, FCs are highly modular systems. As they are stacked, the number of cells per stack and the number of stacks per system can be adjusted easily without significant changes in the efficiency. Thus the efficiency depends much less on the system size than for gas turbines. However, this means that fuel cells do not benefit from upscaling in terms of

efficiency and investment costs, which limits them to rather small units (Tab. 1.5). Even for very sophisticated systems, the round-trip efficiency of a FC storage is below 50 % [68].

The most significant disadvantage of FCs are the high investment costs. Sharaf and Orhan [165] concluded that costs and durability are the main barriers preventing fuel cells from being economically-competitive in the energy market. The latter is with approximately five years of operation significantly shorter than for other energy generation systems. For these reasons, FCs seem to be more suited in the mobility or distributed heat and power sector than for large-scale grid services.

1.4.2 Hydrogen Combustion with Air

The industrial combustion of gaseous fuels as natural gas has been optimized during the last decades and evolved very sophisticated solutions to feature high efficiency while keeping harmful emissions low. The pollutants carbon monoxide (CO), nitrogen oxides (NO_x), and unburned hydrocarbons (UHCs) are by-products of the combustion process and can be reduced to a certain degree. However, the largest greenhouse gas emission is CO₂, which is inherent for the combustion of natural gas and other hydrocarbons, as shown by the global reaction equation of methane (CH₄) (Eq. (1.1)). Here CO₂ and steam are the reaction products. In contrast, the combustion of hydrogen does not generate CO₂ at all. Moreover, CO and UHCs are also avoided due to the missing carbon in the fuel.



Even enrichment of natural gas with H₂ will reduce the CO₂ emissions. However, the combustion of H₂ is very challenging because its characteristics differ much from the hydrocarbons. It has a much higher energy density in terms of mass, but the volumetric one is low. Even more important, H₂ features a strongly increased reactivity and therefore promotes higher flame temperatures and speeds. The temperature affects the emissions and the durability of machine components while the flame speed influences the flame stabilization, which is discussed in Sec. 2.3.1.

An overview of the potentials and challenges of fuel flexibility at industrial-scale gas turbines was given by Lieuwen et al. [117]. The utilization of H₂ in these turbines, which were initially designed for natural gas, was evaluated theoretically by Chiesa et al. [38]. They proposed measures for NO_x reduction and discussed the influence of H₂ on the complete system. Gazzani et al. [67] extended this study by simulating a combined cycle designed explicitly for hydrogen combustion. They compared the implications of two types of combustors, namely diffusive and premixed H₂ combustors. They found that fuel dilution always reduces the combined cycle efficiency. However, premixed hydrogen combustion is much more affected by the H₂ related challenges of the combustion process mentioned above.

Therefore, premixed combustion of H₂ and H₂-rich fuel has been increasingly studied recently. The Sandia National Laboratories investigated the flame stability and the emissions of mixtures with 45 % H₂ in natural gas [157, 159]. H₂ enrichment increases the risk of flashback but improves the lean blow-off limit, enhancing the flammability range. The flashback propensity was studied by Beerer et al. [21], Page et al. [136], and Syred et al. [181]. They also measured turbulent flame speeds at elevated pressures and temperatures. In general, numerous studies showed that fuels with high H₂ contents reveal higher flashback propensity

as well as increased NO_x emissions [56, 119, 179]. Those issues must be solved and, therefore, gas turbine manufacturers such as Siemens [27, 103, 107–110, 200], GE [37, 106, 202], Mitsubishi Heavy Industries [12, 86, 133], Solar Turbines [6] and PSM [29] enhanced their research to adapt conventional combustion systems for fuels with high hydrogen content. Most of the mentioned publications reported problems with the NO_x emissions or the flame stabilization, which limited the H_2 content in the fuel or required dilution. Du Toit et al. [52] gave an overview of the available machines and recent developments. A well-suited system for higher H_2 content is the sequential combustion approach used by Ansaldo Energia's GT36 heavy-duty gas turbine [138]. This two-stage concept utilizes a premixed flame in the first and an auto-ignition flame (see [1]) in the second stage. Ciani et al. [40] and Bothien et al. [25] showed in high-pressure tests that redistributing the fuel from the first to the second stage allows H_2 enrichment up to 70 %_v without changing the combustor hardware. Further enrichment to pure H_2 was achieved by slightly reducing the combustor outlet temperature and, thus, derating the machine.

Despite all challenges, two gas turbines running on pure H_2 were commissioned in 2010 in Fusina, Italy, and in 2018 in Kobe, Japan, for combined heat and power usage. The first avoids high emissions and high flame temperatures by diluting the flame with steam. The latter uses the micro-mix combustion concept, where instead of one big flame, several small diffusion flames are used [65, 183]. In this concept, fuel and air do not mix upstream. Instead, the fuel is directly injected into the combustion chamber. A similar concept was studied by NASA [124] and is known as Lean Direct Injection (LDI) combustor. Due to the multiple small injections, the fuel is assumed to mix upstream of the flame front. Hence, this concept enables the advantages of premixed combustion but without risking flashback.

In the current thesis, both measures, steam dilution and direct fuel injection, were applied to control the flame temperature and stabilization. In contrast to the mentioned studies, the combustor in this thesis runs on pure oxygen instead of air.

1.4.3 Hydrogen Combustion with Oxygen

The combustion of H_2 is not emission-free even though it does not emit CO_2 , CO , or UHCs. The available literature shows that NO_x emissions are a significant concern for the combustion of hydrogen in the power generation sector. The generation for NO_x can be easily avoided by removing the nitrogen (N_2) from the oxidizer, i.e., using pure O_2 instead of air. The global reaction then simplifies to Eq. (1.3).



If the reaction consumes all H_2 and O_2 , steam will be the only product. Steam itself can also be regarded as pollutant emission because it contributes to global warming, especially when emitted in the atmosphere. In stationary power plants, however, most of the steam can be condensed and cycled back.

The usage of pure O_2 as an oxidizer is known as oxyfuel combustion. The first technical application of pure H_2/O_2 flames originates back to the propulsion systems of space rockets. Here the propellants may be in liquid or gaseous form. The reason for using pure O_2 in these applications is the increased reactivity and the lack of air in space.

In 1978 NASA already reported theoretical investigations to the technical and economic feasibility of the combustion of H_2 and O_2 in turbomachinery [199]. Several concepts of integration were evaluated including supplementary steam generation, H_2 -fueled gas turbines, and the Ericsson cycle. In the early eighties, a consortium of research institutes and industry

partners in Germany started a vast research program. The consortium intended to develop a hydrogen-based measure for the supply of control reserves by the generation of steam from H_2 and pure O_2 . The product steam would temporarily increase the power output of steam or nuclear power plants and replace conventional methods of contingency services. Following some thermodynamic and economic studies [178, 197], an H_2/O_2 steam generator (H_2/O_2 -SG) prototype was developed at the Deutsches Zentrum für Luft- und Raumfahrt (DLR). The development phases were described by Sternfeld et al. [177], including drawings of the prototype and the plant integration. In the H_2/O_2 -SG, the reactants were injected into the combustion chamber at their stoichiometric ratio. Feedwater was injected downstream of the reaction zone to cool down the exhaust to the desired temperature and increase the steam output. The prototype was already at an industrial scale and intended to provide a maximum thermal power of 70 MW that was available within a few seconds. Beer et al. [20] defined the system requirements for the plant integration regarding process controlling and combustion efficiency. More insights into both aspects of the project are given in the introduction of Chapter 5. The experimental results of the prototype can be found in [62, 176]. The main problems were to reach the extremely high targeted combustion efficiency and the wall cooling of the combustion chamber [78].

Nevertheless, Haidn et al. [77] summarized that the technical issues are solvable by system optimization and that the system never came to the field testing stage only because of economic reasons. Low fossil fuel prices, as well as a lack of ecological awareness, reduced the motivation to introduce new energy systems. However, they concluded in 2009 that the energy market's situation has changed and that they see a good potential for the application of the H_2/O_2 -SG, especially in emerging energy systems with low grid capacities [77].

In addition to the efforts in Germany, there were similar projects in Japan and Russia. The Japanese World Energy NETwork (WE-NET) project studied the possibilities of an international hydrogen energy network [83, 129]. This project intended to develop a turbine utilizing H_2/O_2 combustion. Three different combustor concepts have been evaluated, as stated in the annual reports [132]: (1) Annular type combustor, (2) can-type combustor with steam dilution after direct H_2/O_2 combustion, and (3) can combustor type with the combustion of H_2 and a steam/ O_2 mixture. These concepts have been tested in a high-pressure combustion test rig at 2.5 MPa and 1700 °C turbine inlet temperature. Hijikata et al. [83] reported stable flames from ignition up to the rated conditions and a residual H_2 and O_2 concentration of 1 % at stoichiometric combustion. In a numeric evaluation of their suitability, concept (1) was rated best because of the high combustion stability and the most homogeneous temperature distribution. However, the results of the tests were not adequately published, and there is a lack of information about the exact flow and flame behavior, the utilized measurement techniques, and its accuracy. It was stated in a future perspective that the reliability of the measurements must be verified and improved. The research activities were discontinued in 1998, but a group at The Institute of Applied Energy, Tokyo, has recently picked up the topic.

On the Russian side, Malysenko et al. [122] reported the design and evaluation of three H_2/O_2 -SG units with 20 – 100 kW (gas-gas), 10 – 18 MW (gas H_2 -liquid O_2), and 25 MW (gas-gas). The design is very similar to the DLR H_2/O_2 -SG and based on rocket combustor technology. Impinging jets of H_2 and O_2 directly combust in the combustion chamber with a downstream dilution of water for vaporization. The dilution mass flow marks around 70 % of the exhaust stream ($\Omega = 2.33$). As in the projects mentioned before, the combustion efficiency (here named completeness or perfectness) plays a crucial role. They gave a target value of 98 % and reported quenching at the wall of the combustion chamber as a critical issue. However, Malysenko et al. [122] mentioned that optimization of the combustion efficiency

might not be the only goal, as non-condensable gases can also be extracted afterwards or non-stoichiometric combustion can be applied. In the end, the plant needs to be economically optimized as a whole to find the best compromise [122]. Successful testing of the 25 MW class has been reported by Malysenko et al. [123] in 2012. They proposed an application of the unit for industries that produce hydrogen as a by-product and are in the need of autonomous production of heat and power. Moreover, it is suitable for grid balancing and accumulation of electric power at nuclear and steam power plants as well as at plants with renewable energy sources and a recuperation coefficient of more than 50 % in the range of 0.1 – 100 MW [123]. A similar device in the order of 200 kW for energy storage purposes was recently presented by Borzenko and Schastlivtsev [24, 156].

In the micro-scale (< 5 kW) range, Guryanov et al. [74] presented a counter-current H_2/O_2 vortex combustion chamber. In their experiments, the combustor was directly supplied by a premixed H_2/O_2 mixture from an electrolyzer. Hence, it was ensured that the mixture is stoichiometric, as the authors stated. The gas was injected into the combustion chamber as a jet in the core of the swirling steam flow. The study investigated mainly geometrical parameters of the fuel nozzle and the combustion chamber. The combustion efficiency was not measured.

1.5 Structure of the Thesis

In the current thesis, a lab-scale burner for H_2/O_2 combustion was developed. In contrast to the projects mentioned before, it is not based on rocket technology but swirl-stabilized, steam-diluted combustion. The purpose of the thesis is to document the entire development process and to characterize the burner's operation. Hence, it delivers essential insights and advises for future combustor developments in the field of H_2 oxyfuel combustion.

The central research questions of the thesis are given in the following bullet points. The overall structure of the thesis is related to these questions.

- Which short- and long-term applications does the H_2 oxyfuel combustion offer? Are they economically competitive?
- Which design changes are necessary to adapt a swirl-stabilized, premixed burner for H_2 oxyfuel combustion?
- Combining swirl-stabilization, i.e., a central recirculation zone in the flow field, and direct fuel and oxidizer injection – how is that influencing the flow field and mixing in the combustion chamber?
- What are the operational boundaries of the burner, especially regarding the degree of steam dilution?
- A crucial requirement of such a combustor is the combustion efficiency. How can that be monitored online, and how is it affected by steam dilution and swirl intensity?

In Chapter 2, two possible applications of the combustor are introduced. First, a short-term application as an H_2/O_2 -SG that provides PCR is analyzed for its economic potential. Second, a long-term application as a combustor in a Graz cycle is thermodynamically investigated by employing an exergy analysis. Additionally, the cycle is extended by an electrolyzer and gas tanks to evaluate the round-trip efficiency of an H_2 oxyfuel combustion-based energy storage system. However, both studies have a limited scope since the work

for this thesis has an experimental focus. The last part of Chapter 2 leads over to the experiments by shortly introducing gas turbine combustion fundamentals and illustrating the development process of the burner used in this thesis.

Chapter 3 presents the results of isothermal flow investigations. It focuses on the influence of the steam dilution and the swirl intensity on the flow and mixing field. In addition to the experiments, supporting computational fluid dynamics (CFD) simulations are performed to obtain models describing certain flow features, e.g., the swirl number.

Subsequently to the isothermal investigations, the operation characteristics of the burner are analyzed in a combustion test rig. Therefore, Chapter 4 introduces the test rig, elaborates the flame behavior, determines operational maps, and analyses the heat transfer from the combustor flow to the walls.

As mentioned before, the combustion efficiency is a crucial parameter for the H_2 oxyfuel combustion. For this reason, Chapter 5 is solely dedicated to this aspect. In the corresponding experiments, a new way of measuring the combustion efficiency was employed, which is based on a tunable diode laser absorption spectroscopy (TDLAS) O_2 sensor. This approach is introduced in Sec. 5.4 and has the advantage of being time-resolved. The necessity of perfectly stoichiometric combustion additionally requires the utilization of a lambda sensor (LS) to control the equivalence ratio. Section 5.5 presents the results of the efficiency measurements with both sensors. The uncertainty quantification is given in the appendix.

Finally, the last chapter concludes the findings of the thesis accompanied by an outlook for improvements and advises on the burner development and the efficiency measurements.

CHAPTER 2

Hydrogen Oxyfuel Combustion: Application and Combustor Design

An extensive literature study on the technical realization of the H_2 oxyfuel combustion is already given in Sec. 1.4.3. Now, the current chapter focuses mainly on two aspects: possible applications of the technology and the development process of the swirl-stabilized H_2/O_2 burner.

The first aspect includes a short-term and a long-term application. Both sections introduce possible integrations to existing or new power systems along with brief studies of economic or thermodynamic characteristics. These studies shall have an introductory character to encourage the extension of research on this topic and justify the experimental characterization of the combustor, which is the main objective of the thesis.

The experimental part is introduced in the third section of this chapter, starting with a general background on gas turbine combustor technology. Further, the section describes the development process of the specific combustor from a fuel-flexible swirl-stabilized natural gas burner to the H_2/O_2 burner investigated in this thesis.

2.1 Short-Term Utilization: Supply of Control Reserves

In the short-term, the adaptation of existing power plants towards H_2 utilization seems to be more feasible than realizing complete new cycles. Therefore, the supply of control reserves as a power plant extension is an exciting alternative and has been the motivation for the development of the H_2/O_2 -SGs at the DLR and in Russia. The next paragraphs give an overview of the literature on these attempts.

In the German project, different approaches of utilizing H_2 for peak-load energy generation were thermodynamically studied. Wojkowsky et al.[197] discussed the thermal efficiency of three possible cycle integrations: a large scale H_2/O_2 -SG connected to a steam turbine, a conventional steam cycle with H_2 fuelling and an open H_2 gas turbine cycle. The first of these was reported to have the highest efficiency. Sternfeld et al.[178] extended this study by introducing an application that uses the H_2/O_2 -SG as a booster for the turbines of a conventional steam cycle. In this approach, the H_2/O_2 -SG temporally boosts the power output of the power plant to provide PCR and SCR. It was shown that the levelized cost of electricity (LCOE) of this option is lower than for the other three cycles. Moreover, the LCOE was prognosticated to be nearly independent of the annual operating hours, which is essential for flexible ancillary services. The study also stated that capital and development costs should be low since all the necessary components for this application were already available, except the H_2/O_2 -SG itself. In 1989, Sternfeld et al.[177] planned its utilization in a 157 MW demonstration plant. To increase the power output by 12 MW,

the H_2/O_2 -SG should generate 70 MW thermal power for start-up and 43 MW for steady operation. The steam was injected upstream of the intermediate-pressure turbine (IP-T) at the same conditions as the steam coming from the boiler reheat stage (40 bar, 788 K). The H_2 combustion was, therefore, only used to increase the mass flow, not the temperature. This approach and further integration concepts were discussed economically in 1995 [176].

In a theoretical analysis, Cicconardi et al. [41] proposed to additionally benefit from the H_2/O_2 -SG's thermal advantages instead of just increasing the mass flow. They claimed that such a device could help to overcome the thermal limits of a conventional reheat stage with its steam temperatures of 820 – 870 K. Hence, they replaced the super-heater and the re-heater in the cycle by a mixer with steam from the H_2/O_2 -SG. With this measure, they were able to increase the power plant efficiency from 36.2 % to 58.8 % and yield an H_2 utilization of 61.8 %. However, the exceptionally high turbine inlet temperatures of up to 2000 K, assumed here, require extensive changes to the original power plant, including new turbine cooling technology. The economic aspects of this approach were not discussed.

The most comprehensive work on the thermodynamic and economic evaluation of this approach is the one work of Stathopoulos et al. [174], which will be summarized and extended in the next subsection.

2.1.1 Thermodynamic evaluation of H_2 -based PCR

Stathopoulos et al. [174] reproduced the model of a super-critical coal power plant cycle from Romeo et al. [151] and extended it with an H_2/O_2 -SG for reheating. They studied three possible integration points: upstream of the HP-T, upstream of the low-pressure turbine (LP-T), or both. Figure 2.1 shows the last integration case with two combustors. The first and the second case were identical but with only the first or second combustor, respectively. All of them were designed to require minimal adaptation to the original cycle. Particularly the steam generation block remained unchanged, which enables a regular base-load operation when no control reserves are activated.

In thermodynamic terms, the study found that the second integration case (LP-T) yields the highest thermal efficiency of $\eta_{\text{th}} = 41.6\%$ and the highest H_2 utilization factor of $\eta_{\text{u}} = 63.6\%$ (Tab. 2.1). The thermal efficiency was calculated as the ratio of the cycle's electric power increase to the heat introduction by the H_2 based on the higher heating value (HHV). The utilization factor additionally referred this value to the Carnot efficiency of 65.3 % (see [174]).

Table 2.1: Thermal efficiency and H_2 utilization factor, taken from [174].

Integration case		HP-T	LP-T	HP-T/LP-T	
Power increase (electric)		2.25	15.30	19.13	MW_{el}
Thermal efficiency (HHV)	η_{th}	19.54	41.62	39.75	%
H_2 utilization factor	η_{u}	29.88	63.64	60.77	%

Besides the economic feasibility of the H_2 -based PCR, which will be discussed in the next subsection, the combustor's thermodynamic properties were of interest for this thesis in order to define the operational requirements. In all three integration cases, the task of the H_2/O_2 -SG is to reheat the steam to a specific temperature before it enters the turbine. For example, in the LP-T case, the temperature of nearly 880 t/h steam had to be increased from 637° to 693°C at 11.8 bar. The H_2/O_2 -SG, therefore, must deliver a well-defined high-temperature steam mass flow. The resulting consumption of the combustor would be 0.9 t/h H_2 and 7.4 t/h O_2 . Accordingly, the steam generated by the H_2/O_2 -SG would make

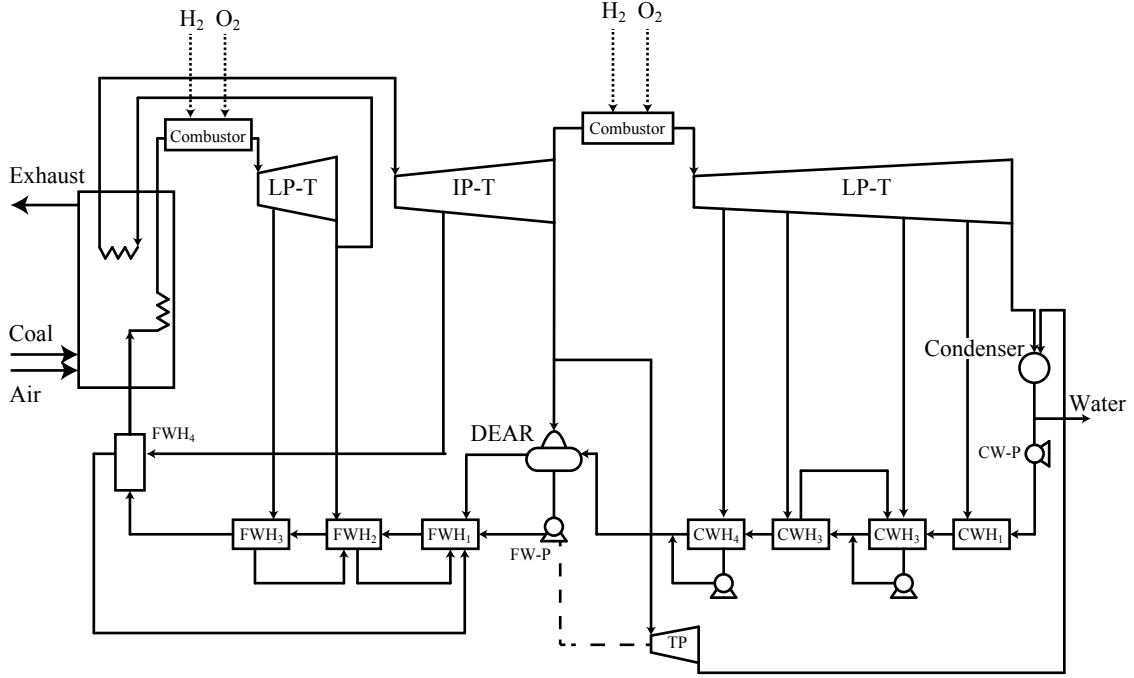


Figure 2.1: Schematic of the third integration case HP-T/LP-T. The figure is taken from [174].

less than 1 % of the total steam delivered to the LP-T. A particular portion of the incoming steam would be used for combustion chamber cooling or flame dilution. Later in the thesis, it will be characterized by the steam dilution ratio Ω (Eq. (2.7)), which would reach $\Omega = 105$ in this case. The combustor operation data of all the three integration cases is shown in Tab 2.2.

Table 2.2: Operation conditions of the combustor for all the three integration cases. In the last case, two H_2/O_2 -SGs are applied.

Integration case	HP-T	LP-T	HP-T/LP-T	
Combustor				
Thermal power (LHV)	9.8	31.3	9.8 / 31.3	MW
Operating pressure	300	11.8	300 / 11.8	bar
Mass flows				
Steam inlet	1260	878	1260 / 882	t/h
H_2 inlet	0.29	0.94	0.29 / 0.94	t/h
O_2 inlet	2.33	7.44	2.33 / 7.47	t/h
Steam dilution Ω	480	105	480 / 105	
Temperatures				
Steam inlet	865	637	865 / 637	K
Reactant inlets	298	298	298 / 298	K
Outlet stream	883	693	883 / 693	K

2.1.2 Economic evaluation of H_2 -based PCR

The H_2 in this approach was assumed to be generated directly on-site by the electrolyzer. The required electricity is either taken from the grid at the industrial electricity price or from

the power plant itself at the LCOE. The offered PCR generates an income of 4000 €/MW weekly (independent from the actual generated power). The economic analysis yielded the net present value (NPV), shown in Fig. 2.2 for both pricing schemes. The jumps in the left graphs were caused by the consumption-based graduation of the electricity price. The figure reveals that integration case two with the first pricing scenario is viable for less than 250 operational hours per year. Otherwise, the NPV gets negative. By applying the on-site LCOE model (pricing scenario two), the maximum number of activation hours increases to 450 h in case two.

A sensitivity study by Stathopoulos et al. [174] showed that the capital costs (including the electrolyzer and gas storage) have much less influence on the NPV than the electricity and PCR prices. The sensitivity of the latter is shown in Fig. 2.3 (left) for a variation of $\pm 30\%$. The PCR prize is governing the profitability of the approach. Hence, it is worth reviewing its long-term development, as done in Fig. 2.3 (left). Since its maximum average value of 4000 €/MW in 2015, taken for the study by Stathopoulos et al. [174], the price has dropped in strong fluctuations to a current value of below 2000 €/MW. Based on this decline, one can expect much lower profitability. For this reason, the economic study was repeated with updated values for this thesis. Besides the PCR prize, the capital costs of

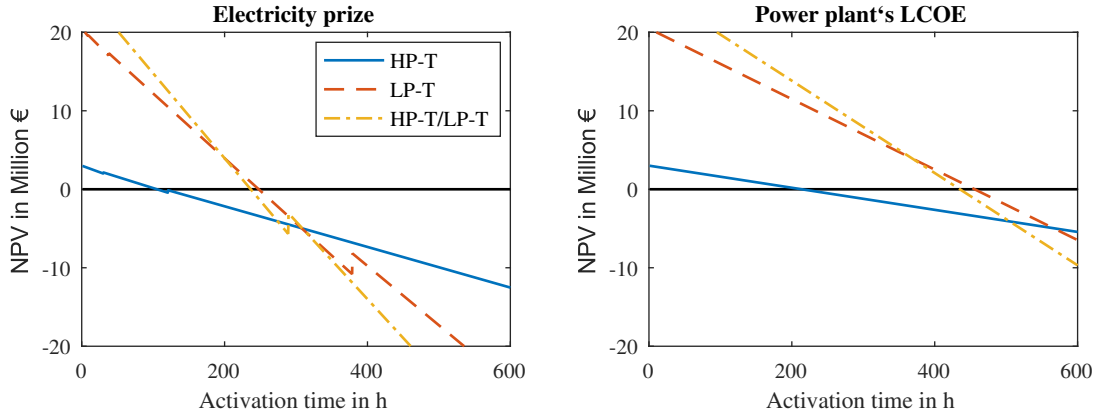


Figure 2.2: NPV of the three integration cases, assuming the price of electricity (left) or the LCOE of the power plant (right) for the costs of the electrolysis. The figure is taken from [174].

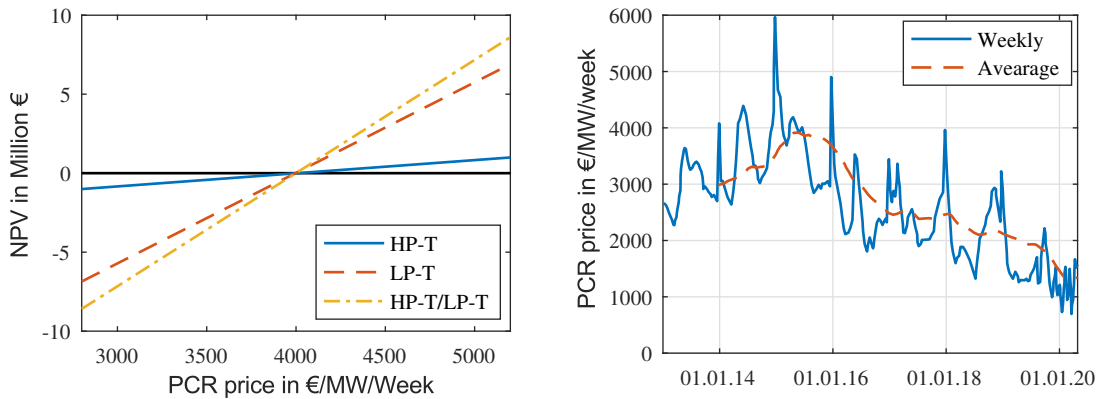


Figure 2.3: Left: Sensitivity of the PCR price on the NPV. Right: Development of the PCR price with a 52-week sliding average (- -) [144].

the electrolyzer [169] and the storage, as well as the LCOE [99], were adjusted. Table 2.3 displays the old and new values. Regarding storage, the costs, reported in the literature, vary strongly in the range of 20-100 €/Nm³ H₂ [191]. Wang et al. [193] reported the prize for H₂ storage in bundles as 36 €/Nm³ or 400 €/kg, respectively. Larger storage can reduce costs. For instance, with salt caverns, the costs can get as low as 60 €/kg [34]. However, the amount of H₂ stored for PCR would not be that large. The current study assumed a prize of 250 €/kg.

Table 2.3: Assumptions of the economic model from the 2007 study [174] and the current study.

	2017 study	Current study	
Specific installation costs			
Electrolysis plant	1100	920	€/kW _{el}
Storage tanks	509	250	€/kg
Combustor	300	300	€/kW _{th}
Operational costs			
Electrolysis fixed operational costs	4	2	%/a
Electrolysis variable costs	0.3	0.3	€/MWh
Storage fixed costs	2	2	%/a
Assumptions for the NPV			
Plant operation life	20	20	years
Discount rate	10	10	%
Prices			
PCR price	4000	2000	€/MW/week
LCOE coal power plant	75	80	€/MWh
LCOE wind power plant	-	55	€/MWh

Re-evaluating the economic model of Stathopoulos et al. [174] yields the NPVs in Fig. 2.4. Even with the better pricing scenario (LCOE) and LP-T integration case, the NPV will only be positive for less than 245 hours of activation, i.e., three activations per day. However, the model makes some conservative assumptions. First, an activation always requires the full PCR capacity for the activation period of 15 minutes. Second, the electrolyzer size is chosen minimal, which means that the electrolyzer runs all the time, except when PCR is activated. Alternatively, one can increase the electrolyzer size, which would allow for a reduction of the storage size. If one changes both sizes simultaneously, it would be advantageous for the NPV, as shown in Fig. 2.5. The reason is that the electrolyzer is switched off during activation and, thus, contributes to the amount of power offered for PCR. As a result, the income of the PCR provision increases with a larger electrolyzer and compensates for its higher capital costs. On the other hand, the smaller storage size reduces capital costs and provides an overall higher NPV. Moreover, a larger electrolyzer gives the operator of the system more freedom to react to the market situation, which would create additional economic potential. However, the dynamics of the market were not covered by the steady model used.

2.1.3 Economic evaluation of H₂-based PCR including a wind farm

The introduction shows that power generation by wind energy is actively expanding in Germany and other industrial countries. It is highly volatile and might amplify the temporal fluctuations of the electricity generation. Hence wind energy should also contribute to the provision of grid services to balance generation and consumption. This section shortly

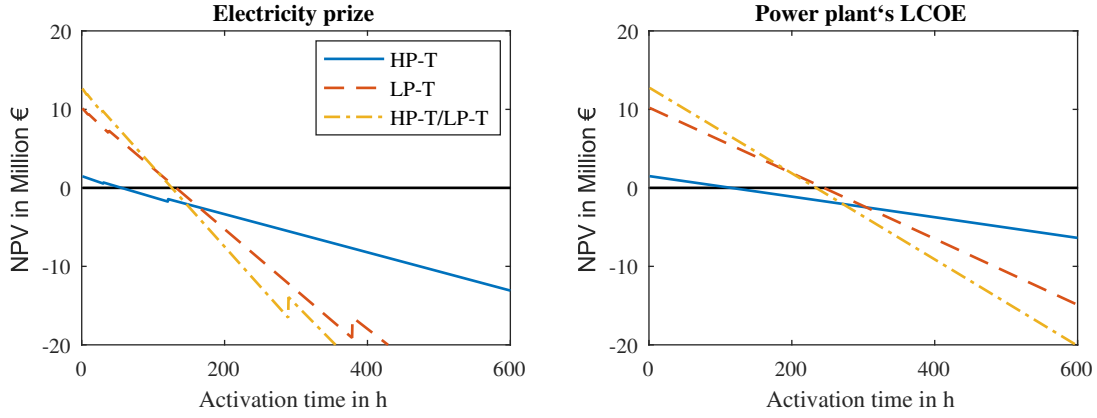


Figure 2.4: NPV of the three integration cases with updated costs (see Tab. 2.3), assuming the price of electricity (left) or the LCOE of the power plant (right) for the costs of the electrolysis.

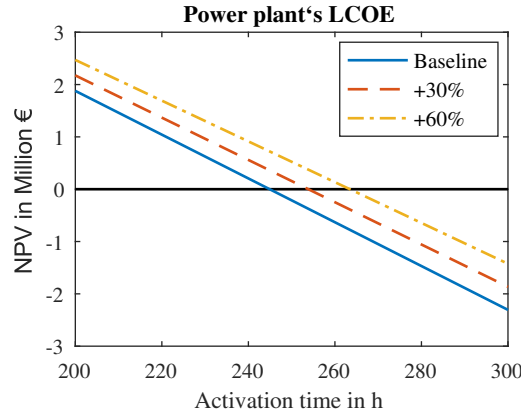


Figure 2.5: Influence of the electrolyzer plant size on the NPV (prizing scenario: LCOE of the power plant; integration case: LP-T). Solid line: Electrolyzer produces continuously H_2 and O_2 based on the daily consumption. Dashed line: 30 % larger electrolyzer and 30 % smaller gas storage. Dash-dotted line: ± 60 %.

evaluates the possibility of integrating wind power in the PCR concept mentioned before.

Time-Resolved Storage Model

The study was based on the time-resolved power output of a 1.8 MW onshore wind turbine, recorded over two years. This way, the fluctuations of wind energy are incorporated into the model. Figure 2.6 gives an insight into three months of generation and an impression of its high volatility. This data was scaled up to the desired size of the wind farm, which is a free input parameter of the model. The model is described in the next paragraphs.

The electricity generated by the wind turbines serves as a power source for the electrolyzer. However, since the wind turbines cannot operate continuously on their nominal power, they should be sized significantly larger than the electrolyzer. Naturally, the usable power of the electrolyzer is limited by its size. If the power output of the wind turbines is above this limit, the excess energy must be fed to the grid. In the model, it is sold at a fixed price of 70€/MWh. Moreover, the electrolyzer switches off, if its load level drops below 10 %.

The second restrictive factor is the storage size. Therefore, the filling level of the storage

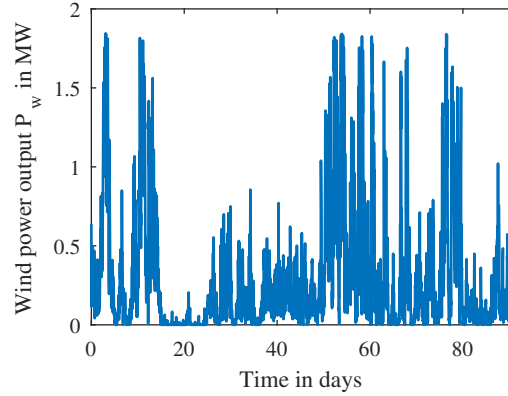


Figure 2.6: Exemplary power generation of an 1.8 MW onshore wind turbine.

is simulated for the two years of available wind power data. The gas production of the electrolyzer charges the storage. It is discharged by the activations, which are, due to a lag of time-resolved data, equally distributed in time according to the total number of activations. If the storage is full, the electrolyzer switches off automatically, and the wind energy is entirely fed to the grid. If the filling level drops below the minimum level (the amount of gas needed for one PCR activation), the electrolyzer will be operated on a 100 % load for one day to refill the storage. If the current power output of the wind park does not suffice for a full load operation, the power difference is balanced by the coal power plant at the LCOE.

Load Simulation of a Reference Case

To better understand the load simulation made in the model, it was conducted for a reference case (Tab. 2.4) and is shown in Fig. 2.7 for a short time sequence. On the left side of the plot, one finds the power generation of the wind turbines as a percentage of the electrolyzer size. The right plot shows the electrolyzer's load level as well as the storage fill level for the same time frame. In the first two days, the wind is powerful, and the turbines generate more energy than consumed by the electrolyzer, which, in turn, runs on 100 % load. The excess

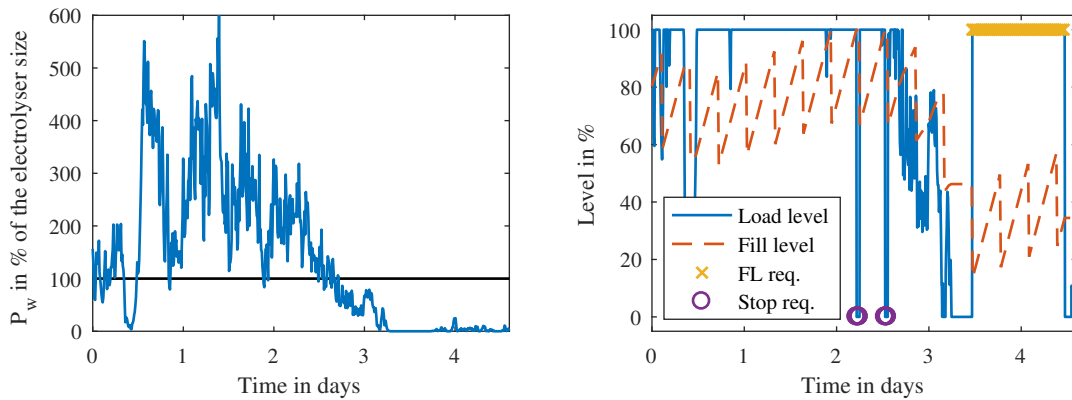


Figure 2.7: Left: Power generation of the wind turbines in relation to the electrolyzer size for the reference case from Tab. 2.4. Right: Load level of the electrolyzer and fill level of the storage for the same case. The yellow crosses mark full load (FL) requests to keep the storage charged. The purple circles mark stop requests due to full storage.

energy is fed to the grid. The storage fill level increases and drops periodically due to the PCR activations. Shortly after the end of the second day, the storage is fully charged and the electrolyzer needs to stop operation, as marked by the circles. From the end of day three on, there is no wind power available anymore. Hence, the storage quickly discharges until the minimum level is reached, and a full load request is triggered. This point is marked by the crosses in the figure. The electrolyzer runs now on full load for one day, even if there is not enough wind power to feed it. The necessary power needs to be taken from the coal power plant. After one day, the electrolyzer switches back to regular operation, i.e., controlled by the available wind power.

The inputs and outputs of the model are summarized in Tab. 2.4, together with their values of a reference case. In contrast to the model of Stathopoulos et al. [174], which features 44 weeks of operation per year, the plant in the current model operates the whole year to match the wind power data. If the earlier model is also extended to a full year, the break-even point, i.e., at which the NPV is zero, would be at 300 activation hours per year. This value is taken for the reference case. The size of the wind power farm is set to 14 MW such that its energy generation matches the total consumption of the electrolyzer over the two years. Consequently, the excess energy of the wind turbines (7000 MWh/a), which is sold to the grid, equals the energy that needs to be added during low wind periods. The NPV of this reference case is EUR 1.33 million and, thus, more profitable than in the model without the wind energy.

Table 2.4: Input and output parameters of the model with values for a reference case.

Input			Output		
Activation time	300	h	Excess energy	6956	MWh/a
Nominal wind farm size	14	MW	Needed energy	6960	MWh/a
Electrolyzer size	2	MW _{el}	NPV	1.33	Million €
Storage size (H ₂ +O ₂)	6.3	t			

Influence of the Wind Farm Size

One can conduct an optimization with the wind farm, electrolyzer, and storage size as parameters and the NPV being the objective. This approach leads to a maximization of the wind farm size while the electrolyzer and storage sizes are set to their lower boundaries. These are the continuous operation for the electrolyzer and twice the gas consumption of one activation for the storage size.

Assuming these minimal sizes of the electrolyzer and the storage, the model yields a positive NPV for a wind farm size of more than 2 MW, as the solid line in Fig. 2.8 (left) shows. In this range, the harvested wind energy is not sufficient to cover the consumption of the electrolyzer. Hence, a large amount of energy must be purchased from the coal power plant or the grid. This needed energy is shown in Fig. 2.8 (right), together with the wind farm's excess energy that naturally is very low for small wind farms. As the wind farm size increases, both curves are approaching each other and meet at 14 MW, where the energy generation of the wind turbines matches the total consumption of the electrolyzer as in the reference case. A further enlargement of the wind farm leads to a nearly linear increase of the NPV because more excess wind energy is sold to the grid.

The electrolyzer needs to run all the time if it has the minimum size, which is optimal in this model. However, the model is quite limited because it assumes constant prices and a regularly distributed activation of the PCR. Alternatively to the minimum size, Fig. 2.8

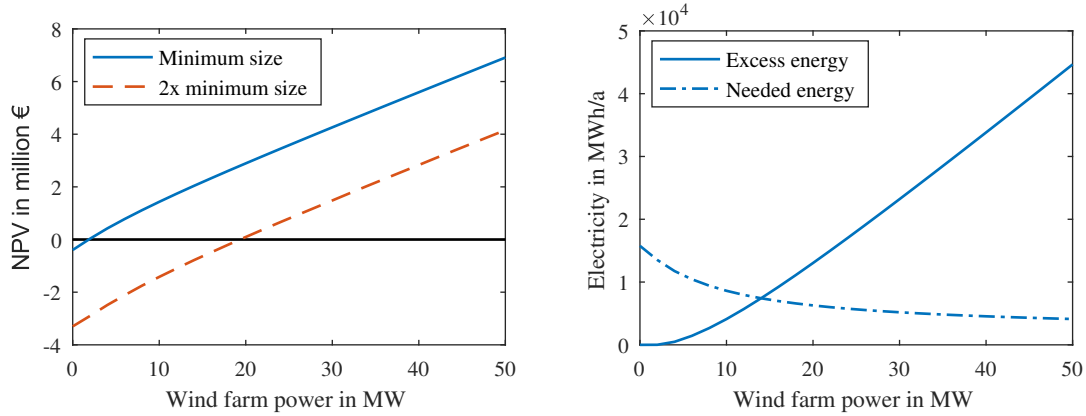


Figure 2.8: Influence of the wind farm size on the NPV (left) and the excess or additionally needed energy (right). The electrolyzer (1.8 MW) and the storage (4.2 t) were set to the minimum size and twice the minimum size (left only).

(left) also shows a variant with a double-sized electrolyzer and storage. The NPV of this version is lower because of the elevated capital costs but offers much more flexibility for the operators. They could adapt the load to the dynamics of the market by using less wind energy for the electrolyzer when the price for electricity feed-in is high or, conversely, fill the storage when the price is low. Hence, strongly fluctuating electricity prices offer the possibility of power arbitrage, potentially increasing the system's profit.

Furthermore, the model does not include the power plants penalty by providing conventional PCR, which would become obsolete with the H₂-based PCR system and enhance its benefits. Theoretically, it is also possible to use the electrolyzer for negative PCR by producing H₂ during positive deviations of the grid frequency. However, Guinot et al. [73] studied this potential and found that under the assumed conditions of the market (France), the H₂ production costs would not reduce when offering PCR.

The brief study performed in this section indicates that there are several opportunities for viably utilizing H₂ storage and combustion for providing PCR. Naturally, a fully dynamic model that incorporates all the temporal effects of the market would be beneficial to judge the profitability of the concept finally. However, this would require a whole project itself and is, thus, out of the scope of this thesis.

2.2 Long-Term Utilization: Zero-Emission Power Plant

On a long-term perspective, H₂ will most likely not remain a niche product for energy storage but become one of the primary energy carriers. Thus, it is necessary to evolve technologies for large-scale H₂ energy utilization. They have to be ecological and, at the same time, highly efficient because of the significant H₂ production costs. In this context, the oxyfuel combustion of H₂ offers the great opportunity of being emission-free and easily scalable. However, at the current stage, the task is to find the optimal thermodynamic cycle to serve oxyfuel combustion.

The idea of oxyfuel cycles stems from the attempt to capture and store the CO₂ emitted by fossil fuel combustion. Kanniche et al. [95] provided an overview of different ways of CO₂ capture in thermal power plants, including oxy-combustion. Possible cycles designed for this kind of combustion are the Allam [7] and the Graz cycle [92], which utilize fossil fuels, pure O₂, and CO₂ or steam as the working fluid. The advantage of these cycles is that the exhaust

consists of almost pure CO_2 and steam, which simplifies CO_2 capturing. Moreover, they feature an enhanced thermal efficiency and lower costs than conventional steam cycles with CO_2 capturing [155]. Besides the fossil-fired version, Jericha and Fesharaki [90] proposed an H_2 -fired Graz cycle. Furthermore, they extended it with an FC in addition to the H_2/O_2 combustor [91]. In general, there is various literature on the Graz cycle, optimizing its working points, analyzing its part-load and off-design performance, or comparing it to other cycles [127, 153, 154, 188].

Alternatively to the Graz cycle, a Rankine cycle can be modified using direct instead of external combustion. In these cycles, the water is boiled in a heat recovery steam generator (HRSG) and super-heated by multiple H_2/O_2 combustors between the turbine stages. Two variants are the Toshiba [63] and the Westinghouse [17] cycle. Both were compared to the Graz cycle by Milewski [126]. Soufi et al. performed an exergy analysis of an enhanced version of the Toshiba cycle, called the New Rankine cycle, and found an exergetic efficiency of 63.6 %. However, this value might be overestimated due to extremely high temperatures in the HRSG. Additionally, the H_2/O_2 Rankine cycles use steam turbines with an inlet temperature of 1973 K, which is about 200 K higher than for the Graz cycle and not feasible today.

In the project underlying this thesis, a master thesis was carried out based on the H_2 -based Graz cycle of Sanz et al. [153]. The work includes an exergy analysis of the cycle, as well as a combination of the cycle with an H_2 storage system [54]. The following section will shortly summarize these investigations and highlight the essential outcomes. For more details on the methods and settings, please refer to Dybe [53].

2.2.1 Thermodynamic Analysis of the Graz Cycle

The Graz cycle consists of a topping Brayton cycle, shown as the solid red line in Fig. 2.9, and a bottoming cycle (dashed blue). Both cycles have steam as its working medium and share the H_2/O_2 combustor, the high-temperature turbine (HT-T), and the HRSG. Hence the cycles run in a semi-closed manner. The bottoming cycle features a deaerator, extracting non-condensable gases from the condensed water. These gases can arise if the combustion of H_2 and O_2 is incomplete, or the reactants are in a non-stoichiometric ratio. However, the steam does not condensate in the topping cycle but is recirculated to the combustion chamber. Hence, non-condensable gases can accumulate over time in the topping cycle if the equivalence ratio is not adequately controlled. A suited control device is part of the experimental studies of this thesis and discussed in Chapter 5.

Unlike the short-term application (PCR system), the Graz cycle requires an HT-T with an inlet temperature of 1773 K, which is not available for steam turbines yet. However, in gas turbines, such turbine inlet temperatures are already feasible, and 2000 K are under development [203]. In the long-term, it can be expected that HT-T for steam will also be applicable. The thermal requirements for the steam generated in the combustion chamber of the Graz cycle are given in Tab. 2.5. They differ clearly from those of the H_2/O_2 -SG in Tab. 2.2. Besides the higher outlet temperature, the thermal power is one order of magnitude larger. At the same time, the cycle offers much less steam for dilution of the flame and cooling of the combustion chamber. Hence, the steam dilution ratio reduces to $\Omega = 4$, which lies in the range experimentally investigated in this thesis.

The thermodynamic analysis of the H_2 -fueled Graz cycle yielded an outstanding thermal efficiency of 70.35 % [153] and a net efficiency of 66.1 % [54] to 68.4 % [153]. A second-law analysis extended these results to obtain the exergetic efficiency of the cycle. The exergy of the fuel/oxidizer mixture corresponds to the work, which can be gained from

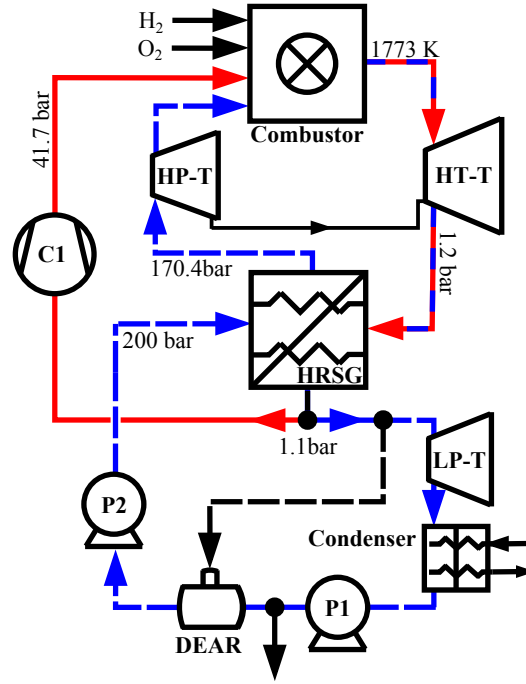


Figure 2.9: H₂-fired Graz cycle [54] with a topping Brayton (red solid) and a bottoming (blue dashed) cycle.

Table 2.5: Operation conditions of the combustor in the Graz cycle.

Integration case	Graz cycle	
Combustor		
Thermal power	296	MW
Operating pressure	41.7	bar
Pressure loss	4	%
Mass flows		
Steam inlet	317	t/h
H ₂ inlet	9.1	t/h
O ₂ inlet	72.2	t/h
Steam dilution Ω	3.9	
Temperatures		
Steam inlet	800 / 630	K
Reactant inlets	288	K
Outlet stream	1773	K

bringing it into a physical, chemical, and thermodynamic equilibrium with the ambient environment. Performing such conversion under non-ideal conditions, i.e., in a machine, generates entropy and exergy losses. The latter are proportional to the entropy production due to irreversibilities. Hence the exergetic efficiency is defined as one minus the ratio of the system's exergy losses to the reactant's exergy. In the case of the Graz cycle, the study presented an exergetic efficiency of $\varepsilon_{Gr} = 64\%$. For comparison, the exergetic efficiencies of a state-of-the-art gas turbine or a combined cycle can be estimated to 32% and 46%, respectively [46, 85].

The advantage of the exergetic over the energetic analysis is that the exergy losses can be

directly assigned to the cycle's components. The study of the Graz cycle showed that the most exergy is destroyed by the combustor – almost two-thirds of the total exergy destruction. This finding highlights the critical role of the combustor in the system. Influencing factors are the type of fuel, the equivalence ratio, and the degree of preheating of the combustor feed-in [198]. However, the utilization of stoichiometric H_2 is already optimal, and the steam's high inlet temperatures reduce the losses. Preheating of the reactants is also reported to decrease the losses [170]. This measure was partly realized in the experimental combustor of this thesis (Sec. 2.3.4 and Appendix A).

Further significant sources of exergy loss in the Graz cycle were found to be the turbines and the HRSG. The water pumps, on the other hand, generated negligible exergy loss compared to the steam compressor. It, thus, justifies the approach of the semi-closed cycle featuring a condenser and compressing liquid water.

2.2.2 The Graz Cycle as Part of an Energy Storage System

Another important question is the efficiency of an H_2 energy storage system, based on the H_2 -fueled Graz cycle. Figure 2.10 outlines such a system, including the electrolyzer, the storage tanks, and two gas compressors. The latter needed to be implemented if the electrolyzer's outlet pressure was less than the combustor's operating pressure. In the study, the compressors increased the pressure from the storage pressure, i.e., the electrolyzer outlet pressure, to 50 bar. Subsequently, control valves reduced it to the operating pressure of the combustor.

The list of electrolyzers with different operating pressures in Tab. 2.6 indicates that the efficiency of the electrolysis declines with increasing pressure. On the other hand, higher storage pressure reduces the necessary work of the gas compressors. Figure 2.11 (left) displays this positive effect of the storage pressure on the energetic and exergetic efficiency. However, the effect is quite weak, especially in terms of the exergy. The reason is that the increase in compressor work also raises the temperature of the reactants, which reduces the exergy losses of the combustion chamber and, therefore, partly compensates the compressor's losses. In fact, the compressors cause only a loss of 0.9 % of the exergy input at 10 bar storage pressure.

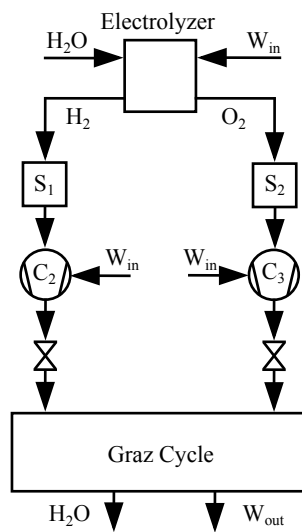


Figure 2.10: Energy storage system featuring a H_2 -fired Graz cycle [54].

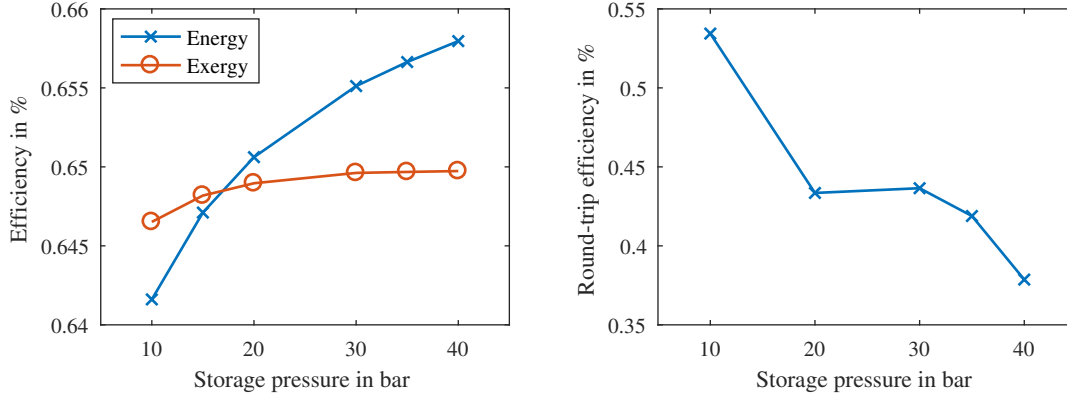


Figure 2.11: Energetic and exergetic efficiency of the Graz cycle with gas pressurization to 50 bar (left) and the round-trip efficiency of the energy storage system (right) [54].

Table 2.6: Commercial electrolyzers with their outlet pressures, the thermal efficiency and the exergetic efficiency [54].

Company	Name	Type	Pressure in bar	η_{EL} in %	ε_{EL} in %
Sunfire	SF 150/30	SOEC	10	83.29	84.58
ITM Power	Hgas	PEM	20	66.63	68.38
H-TEC Systems	EL 450	PEM	30	66.63	68.79
AREVA H ₂ Gen	ELYTE	PEM	35	63.79	66.02
iGas Energy	iEL 30-300	PEM	40	57.55	59.67

For the overall storage system, the lower efficiency of the high-pressure electrolyzers was found to dominate the round-trip efficiency $\Sigma W_{in}/W_{out}$, as shown in Fig. 2.11 (right). While it is below 40 % at 40 bar storage pressure, it grows to 53.4 % at 10 bar. With this high round-trip efficiency, the system would be competitive to the storage systems discussed in Sec. 1.3 because it would additionally offer the advantages of long-time and large-scale storage.

The Sections 2.1 and 2.2 presented two possible applications of the H₂ oxyfuel combustion. The combustor requirements for both were defined in Tab. 2.2 and 2.5, respectively. The next section will line out the development process of such a combustor, i.e., the burner used in the experimental part of this thesis.

2.3 Burner Development for a Hydrogen Oxyfuel Combustor

Successful attempts to realize the H₂ oxyfuel combustion on the machine level have already been made, as described in section 1.4.3. The past projects in Germany and Russia build upon adapting burners from rocket technologies. At these combustors, H₂ and O₂ were injected into a water-cooled combustion chamber, where they combusted at extremely high temperatures. Downstream of the flame, cooling water was injected. It evaporated and, thus, cooled the exhaust stream to the intended exit temperature. These systems had the advantages of easy scalability in terms of thermal power and instant availability of the hot steam [62, 123]. They also could be operated as stand-alone systems because they only required the reactants, H₂ and O₂, and cooling water as supply.

On the other hand, this combustion approach also showed certain disadvantages. Due

to the high adiabatic flame temperature, the combustion chamber walls and the burner elements needed careful cooling to avoid prompt fatigue. However, if the reactants got in contact with the cooled wall before combustion, the reaction could have been quenched. The same applied to the water injection, which could also lead to quenching if the injection rate is too large. Thus the combustors needed a sophisticated design and precise manufacturing to yield sufficient combustion efficiency [78].

In contrast to these attempts, the current project uses the concept of swirl-stabilized combustion, which is commonly spread in state-of-the-art gas turbines. For this purpose, an experimentally elaborated, generic lab-scale H_2 /air burner was adapted for H_2/O_2 combustion.

The next subsection gives a short introduction to the fundamentals of gas turbine combustion with the most important characteristics highlighted in bold letters. A description of the generic burners, which were used as a basis for the development, follows. The H_2/O_2 burner itself was developed in two steps. The experimental findings from the preliminary version will be shortly summarized in Sec. 2.3.3, while the principle investigations of this thesis will focus on the final version of the burner, which is introduced in Sec. 2.3.4.

2.3.1 Fundamentals of Gas Turbine Combustion

The combustor is considered as the heart of a gas turbine. Here, the fuel's chemical energy is released and converted to heat and kinetic energy, which drives the turbine. Sattelmayer [114] summarized the most important aspects of the combustion chamber. Since the thermal efficiency of the process is proportional to the turbine inlet temperature, the working fluid needs to be heated up as much as possible by the combustion system. It must be designed to completely convert the fuel and simultaneously retain the pressure from the compressor. The **pressure loss** should be limited to 3-4 % [114].

A major issue of gas turbine combustion is the generation of polluting **emissions**, i.e., UHCs, CO, CO_2 , and NO_x . While the generation of the greenhouse gas CO_2 is inherent when burning natural gas, the emission of the other pollutants must be reduced to a minimum. Especially the legal requirements for NO_x emissions have become much stricter in recent years. These new requirements led to the introduction of more and more complex combustion systems. However, the combustor still needs to be robust. The flame must be continuously in stable operation, i.e., combustion phenomena like **blow-off**, **flashback**, and **thermoacoustic instabilities** must be prevented. Such events can cause significant damage to the machine and lead to expensive outages.

Non-premixed and Premixed Flames

In particular, the need for low emission combustion systems has introduced a transition to the gas turbine combustor technology. In the beginning, non-premixed (also known as diffusion) flames were commonly used in gas turbines. In this type of flames, the fuel is injected directly into the combustion chamber, where it mixes with the O_2 -containing atmosphere by diffusion. A sufficient fraction of O_2 and fuel must be present to maintain the flame. If the flame is supposed to consume all the reactants, fuel and oxidizer must be in a so-called **stoichiometric ratio**. For instance, the oxidation of H_2 is stoichiometric when the global reaction is represented by the equation



It reads: 2 Mol H_2 reacts with 1 Mol O_2 to 2 Mol H_2O . Accordingly, the stoichiometric ratio of H_2 to O_2 is two in terms of moles, respectively 0.126 in terms of mass (Eq. (5.9)). The **equivalence ratio** ϕ compares the actual fuel/ O_2 ratio to its stoichiometric value:

$$\phi = \frac{\frac{m_{\text{fuel}}}{m_{\text{O}_2}}}{\left(\frac{m_{\text{fuel}}}{m_{\text{O}_2}}\right)_{\text{st}}} \quad (2.2)$$

Depending on the equivalence ratio, flames can be separated into **lean** ($\phi < 1$), **stoichiometric** ($\phi = 1$) and **rich** ($\phi > 1$) flames.

In the case of a **non-premixed flame**, the fuel penetrates into the O_2 containing air. Typical profiles of the species and temperature of a non-premixed flame are given in Fig. 2.12 (left). The profiles were simulated in a 1D reactor using *Cantera*. The fuel mole fraction is constant on one side and zero on the other, the oxidizer fraction vice versa. Both profiles overlap in the center, forming a field of different mixture fractions or equivalence ratios, respectively. Thus, the flame covers a large range from rich to lean conditions. The flame front, which is characterized by high luminescence, is fixed close to the stoichiometric ratio ($\phi=1$) [194]. Fixed by the mixture fraction field, the flame does not propagate but stabilizes itself, which is the main advantage of non-premixed flames. However, this comes in line with large emissions due to the fuel-rich regions of the flame. Therefore, modern gas turbine combustors mostly utilize premixed flames, at least for baseload operation.

In a **premixed flame**, fuel and oxidizer are mixed upstream of the reaction zone. Hence their profiles in Fig. 2.12 (right) are both non-zero on the fresh gas side. Since their mixture ratio is variable but constant, the equivalence ratio can be set to any arbitrary value above the lean blow-off limit. In particular, premixing enables the operation of lean low-emission flames. In this case, there is still oxidizer on the burned gas side, as in Fig. 2.12 (right). The disadvantage of premixed flames is that they are not stabilized by the mixing field anymore. Hence, the flame propagation needs to be adequately controlled, which is explained by the following example.

Stabilization of a Laminar Premixed Flame

Imagine a straight tube filled with combustible gas, as shown in Fig. 2.13. If the gas is ignited at the right side of the tube, it will propagate with a certain velocity, i.e., the **laminar**

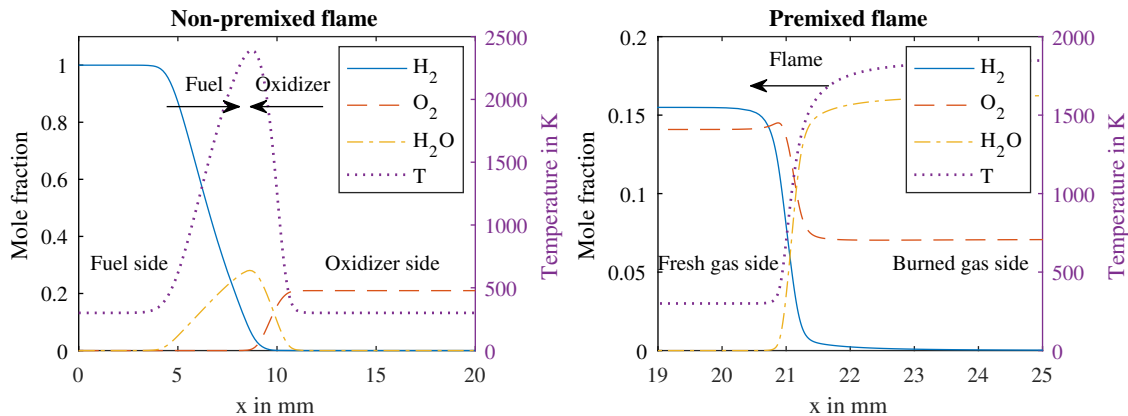


Figure 2.12: Species and temperature profiles from *Cantera* simulations of an opposed-flow diffusion H_2 flame (left) and a freely-propagating, lean premixed H_2 flat flame (right).

flame speed s_L , through the tube until it reaches the left side. The laminar flame speed is a property of a particular fuel that depends on its thermodynamic state. If now a laminar flow is imposed on the gas against the direction of propagation of the flame, the system can turn into three different states.

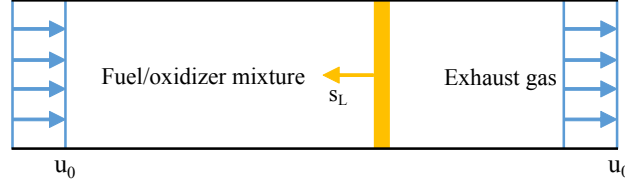


Figure 2.13: Propagation of a laminar premixed flame in a tube.

1. If the flow velocity is less than the flame speed, i.e., $u_0 < s_L$, the flame will still propagate upstream and reach the end of the tube and thus the fuel injection point. This case is named **flashback** and can cause essential damage to upstream components of the combustion system. Hence it needs to be prevented when operating a flame under premixed conditions.
2. If the flow velocity is larger than the flame speed, i.e., $u_0 > s_L$, the flame will be convected against its direction of propagation. In this case, the flame is blown away from its intended position and will possibly be extinguished. Accordingly, this case is called **blow-off**.
3. If the flow velocity matches the flame speed, i.e., $u_0 = s_L$, the flame stabilizes at a fixed position, which is the intended case in a usual combustion system. However, the flame is still not stable against flame speed perturbations, which can be caused by fluctuations of the temperature, pressure, or equivalence ratio.

Stabilization of a Turbulent Premixed Flame

In large scale combustion systems, as mostly used in gas turbines, the flow is not laminar but highly turbulent. The flame front is therefore folded and underlies strain, which significantly accelerates the flame. Thus, the flame speed is no longer a function of the fuel and its thermodynamic state alone but also depends on the flow properties itself, especially on the degree and structure of the turbulence. A correlation to estimate the **turbulent flame speed** is given by

$$\frac{s_T}{s_L} = 1 + \alpha \left(\frac{u_{RMS}}{s_L} \right)^n \quad (2.3)$$

where s_L and s_T are the flame speeds, u_{RMS} is the turbulence intensity, and α and n are constants close to unity [141]. Accordingly, it is impossible to fix the flame position in a simple tube (Fig. 2.13) because of the requirement to set the correct flow velocity continually. Instead, a shear layer must be established, defining a region with different flow velocities that capture the possible flame speeds. The simplest example would be a diffuser flow, as shown in Fig. 2.14. Here the flow underlies an axial velocity gradient, where the velocity upstream is higher and downstream lower than the turbulent flame speed, i.e., $u_1 > s_T > u_2$. As long this condition holds, the flame cannot escape the tube, even if the actual flame speed is fluctuating.

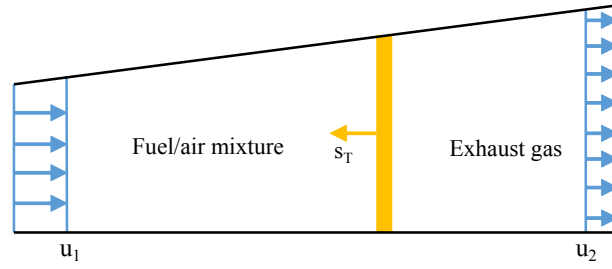


Figure 2.14: Propagation of a laminar premixed flame in a diffuser.

The flame in a gas turbine combustion chamber must be ignited once by external energy and is self-sustaining afterward. The incoming fuel/air mixture is ignited by the heat of the reaction products transported upstream by molecular and turbulent diffusion. Hence, the rate of diffusion limits the propagation speed of the flame. Consequently, the flow velocity must also be limited to avoid blow-off, confining the mass flow and energy conversion density [114]. In order to increase the mass flow, the heat transport must be enhanced, which can be done to a certain degree by increasing the intensity of the turbulence or by creating a recirculation of burned gases in the flow field. The latter is very effective because, in addition to the diffusion, the heat is convected by the backflow, a local flow region that points against the main flow direction. Figure 2.15 gives three technical solutions to establish such a recirculation zone. These are a bluff body (left), a backward-facing step (middle), and a swirl-stabilized (right) combustor.

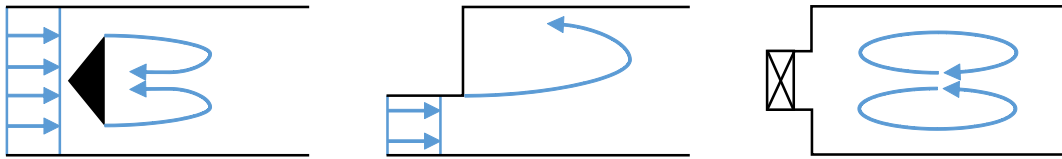


Figure 2.15: Combustor types that create a recirculation zone in the flow field.

In the latter case, the recirculation is created by implying a steady swirling motion onto the incoming flow. This is done by the swirl generator, symbolized by the cross in Fig. 2.15 (right). At the entrance to the combustion chamber, the geometry has an area jump. The flow is less confined, and the swirling jet is opening itself outwards due to centrifugal forces. This spread of the vortex goes along with a pressure minimum at the central axis, forcing the core flow to go against a positive axial pressure gradient. If this gradient gets too large, i.e., the swirl intensity is sufficiently high, the core flow changes its direction and forms a recirculation zone. The advantage of this flame stabilization technique is that the flow field recirculates a large amount of hot gas back to the combustion chamber entrance, where it preheats the incoming reactants [114]. It, therefore, provides excellent properties in thermal mixing. Moreover, the recirculation zone is located in the center of the combustion chamber and does not attach to any wall or body, reducing high-temperature fatigue or flame quenching. For these reasons, the swirl-stabilization is used in most of the modern gas turbine combustors [47, 55, 120]. Therefore, this concept was also applied to the burner in the current thesis.

2.3.2 Generic Model Burner for Fuel-Flexible Combustion

The development of the H_2/O_2 combustor was conducted in two steps with a preliminary and a final version. Their design was derived from the extensive experience on H_2 and

steam-diluted combustion gained with two generic swirl-stabilized burners, namely the movable block (MB) burner and the HP burner, which are briefly introduced in this section.

The preliminary design of the H_2/O_2 burner was developed based on the MB burner. In this burner, the combustion air was guided through an adjustable swirl generator, consisting of fixed and movable blocks. By rotating the movable blocks, the area ratio between the axial and the tangential entrances of the swirl generator was varied. Thus, the flow's tangential momentum could be increased or decreased to affect the swirl intensity. The annular duct between the swirl generator and the combustion chamber is named the mixing tube. Here, the fuel was injected axially at the bottom and mixed with the swirling air. In the center of the mixing tube, a solid centerbody was placed.

The MB burner was extensively investigated at the Chair of Fluid Mechanics at Technische Universität Berlin. Terhaar et al. [185] applied the MB burner with methane and H_2 flames in dry and steam-diluted conditions. In further work [184], they focused on coherent structures in the flow field and the corresponding flame dynamics [186, 187]. The thermoacoustic behavior of the MB burner was investigated by Schimek et al. [160, 161], Ćosić [44], and Oberleithner et al. [135]. Krueger et al. [101] conducted Large Eddy Simulations (LES) of the burner's isothermal flow field and of steam-diluted H_2 flames.

For the combustion of the high reactivity fuel H_2 , a centerbody-free burner design is favorable because the flame anchoring in the vicinity of the centerbody requires extensive cooling to avoid thermal damage [184]. On the other hand, a centerbody-free mixing tube increases the risk of flashback. The flashback mechanisms in such combustors are discussed by Lieuwen et al. [118]. Five different flashback types can be defined for swirl-stabilized combustors [146], from which two are of particular importance for the centerbody-free design: the flashback in the core flow and the combustion-induced vortex breakdown, which was first identified by Fritz et al. [61]. To avoid these types of flashbacks, reducing the velocity deficit on the central axis of the mixing tube would be beneficial. The vortex breakdown should be located well downstream in the combustion chamber. Hence, Burmberger and Sattelmeyer [32] proposed to inject a low axial air momentum to the mixing tube. Reichel et al. [148] showed that a sufficient amount of axial air injection enables the flashback safe operation of dry H_2 flames.

The concept of axial air injection was realized in a second model combustor, the HP burner. The centerbody was removed and the swirl generator changed from the moveable blocks to slots, which guide the air tangentially into the mixing tube. In order to increase the swirl intensity, the slots can be partially blocked, which increases the velocity and, therefore, the tangential momentum. A numerical parameter study of the influence of the blocking on the isothermal flow field was conducted by Tanneberger et al. [182]. This type of swirl generator was utilized in the final version of the H_2/O_2 burner. More experimental and numerical results on the HP burner can be found by Reichel et al. [145–147], Kuhn et al. [102], Stathopoulos et al. [173], and Mira et al. [128].

2.3.3 H_2/O_2 Burner: Preliminary Version

Burner Design

The first preliminary version of the H_2/O_2 burner is shown in Fig. 2.16. In contrast to the premixed H_2 combustion conducted with the MB and HP burner, the flow entered the combustion chamber non-premixed. This measure has the advantage that the flame cannot flashback into the premixing section. The working medium in the H_2/O_2 burner was steam instead of air. However, since steam does not contain an oxidizer, additional O_2 had to be

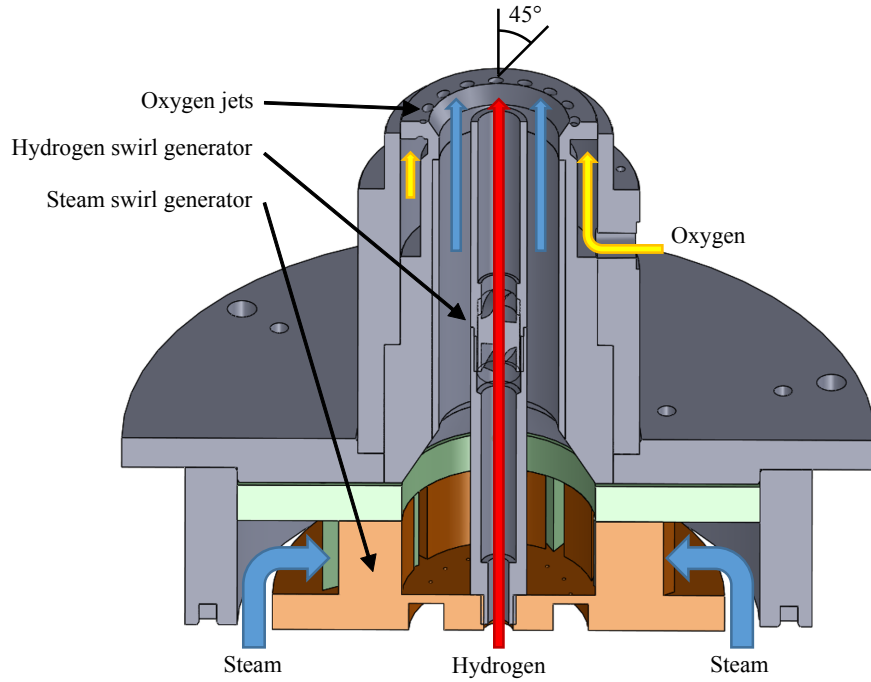


Figure 2.16: Schematic of the H_2/O_2 burner' preliminary version.

injected. In total, there were three different components to be injected into the combustion chamber. The largest share by mass was the steam, which diluted and cooled down the flame. The steam was guided the same way through the swirl generator as the air in the MB burner. The centerbody in the middle of the annular steam duct housed the H_2 nozzle, while the 16 single O_2 jets were placed circumferentially around the annular steam duct. With this layout, the steam flow separated the reactants from each other. Thus they could not get in contact and react close to the burner outlet, avoiding thermal damage to the combustor.

In a first assessment of the flow field, it was found that the swirl intensity of the steam flow was not sufficient to create a suitable flow field in the combustion chamber. For that reason, an axial swirl generator was installed in the H_2 nozzle. Additionally, the O_2 jets were tilted about 45° in the tangential direction. Hence all three components entered the combustion chamber in a swirling motion with the same direction of rotation. A 200 mm diameter and 300 mm length combustion chamber, made of fused silica, allowed optical access to the flame during the experiments.

Combustion Experiments

At first, the combustor was operated with a steam-diluted H_2 /air flame. Thereby, the super-heated steam and the air were mixed upstream of the burner at a temperature of 473 K. After successful ignition, the air content was reduced step by step, while an equivalent mass of pure O_2 was injected into the combustion chamber to hold the equivalence ratio constant ($\phi = 1$). The thermal power was thereby kept at approximately 58 kW, which corresponds to an H_2 mass flow of 1.75 kg/h. The results of the flame measurements [162] are shortly summarized in the following.

O₂ Enrichment and Steam Dilution Ratio

The fraction of O₂ injected directly into the combustion chamber is described with the O₂ enrichment ratio that is defined as

$$\Pi = \frac{\dot{m}_{\text{O}_2}}{\dot{m}_{\text{air}} \cdot 0.23 + \dot{m}_{\text{O}_2}}. \quad (2.4)$$

It was varied in the experiments in four steps from $\Pi = 0$, i.e., no O₂ enrichment, to $\Pi = 1$, i.e., pure oxyfuel combustion. Additionally, the amount of steam dilution was varied in order to influence the adiabatic flame temperature. This parameter is described by the steam dilution ratio

$$\Omega^* = \frac{\dot{m}_{\text{H}_2\text{O}}}{\dot{m}_{\text{air}} \cdot 0.23 + \dot{m}_{\text{O}_2} + \dot{m}_{\text{H}_2}}, \quad (2.5)$$

where $\dot{m}_{\text{H}_2\text{O}}$ is the mass flow of the steam dilution. Generally, it gives the mass ratio of the steam that is used as a diluent to the steam generated by combustion. In the case of $\Pi < 1$, i.e., if air is present in the combustion process, the flame is additionally diluted by the nitrogen contained in the air. Therefore a second dilution ratio is defined to compromise also the cooling effect of the nitrogen

$$\Omega^+ = \frac{\dot{m}_{\text{H}_2\text{O}} + 0.77 \cdot \dot{m}_{\text{air}}}{\dot{m}_{\text{air}} \cdot 0.23 + \dot{m}_{\text{O}_2} + \dot{m}_{\text{H}_2}}. \quad (2.6)$$

Later in the thesis, the flame was solely operated on pure O₂ ($\Pi = 1$), which simplifies the definition of the steam dilution ratio to Eq. (2.7). In this form, Ω will be the primary parameter for comparing the results throughout the thesis.

$$\Omega = \frac{\dot{m}_{\text{H}_2\text{O}}}{\dot{m}_{\text{O}_2} + \dot{m}_{\text{H}_2}} \quad (2.7)$$

Flame Characteristics

In the initial combustion experiments, the combustor was investigated in the range of $\Omega^* = 1.2$ -3.8. Stable combustion was found in every case observed. Figure 2.17 shows the OH* chemiluminescence recordings of multiple measurement points across this range. Each row shows a certain level of O₂ enrichment Π , ranging from 0 to 1, i.e., from air to pure oxyfuel combustion. In the first column, the flames of the lowest applied steam dilution ratio are depicted. They vary, depending on Π from $\Omega^* = 1.17$ to 1.94. The second and the third columns show flames with a fixed steam dilution ratio of $\Omega^* = 2.45$ and 3.76, respectively.

The images of the heat release zone indicate that the flame had sufficient distance from the burner exit to prevent thermal damage over the whole operational range. All flames revealed the typical conical shape of a swirl-stabilized flame [162]. By reading the plots from left to right, one can see how the steam dilution affected the intensity of the flame's OH* chemiluminescence in a reducing manner. The flame was convected downstream and spreads in the radial direction. This effect could be explained by the reduced flame speed under steam-diluted conditions [71]. At increasing amounts of O₂ enrichment, the effect of the steam dilution degraded significantly. The strong chemiluminescence at $\Pi = 1$ and $\Omega^* = 3.76$ indicated that the cold blow-off limit of the oxyfuel flame was at much higher steam dilution ratios than for the air flame. However, due to the steam generator's limited mass flow, none of the flames reached the blow-off boundary in the experiments.

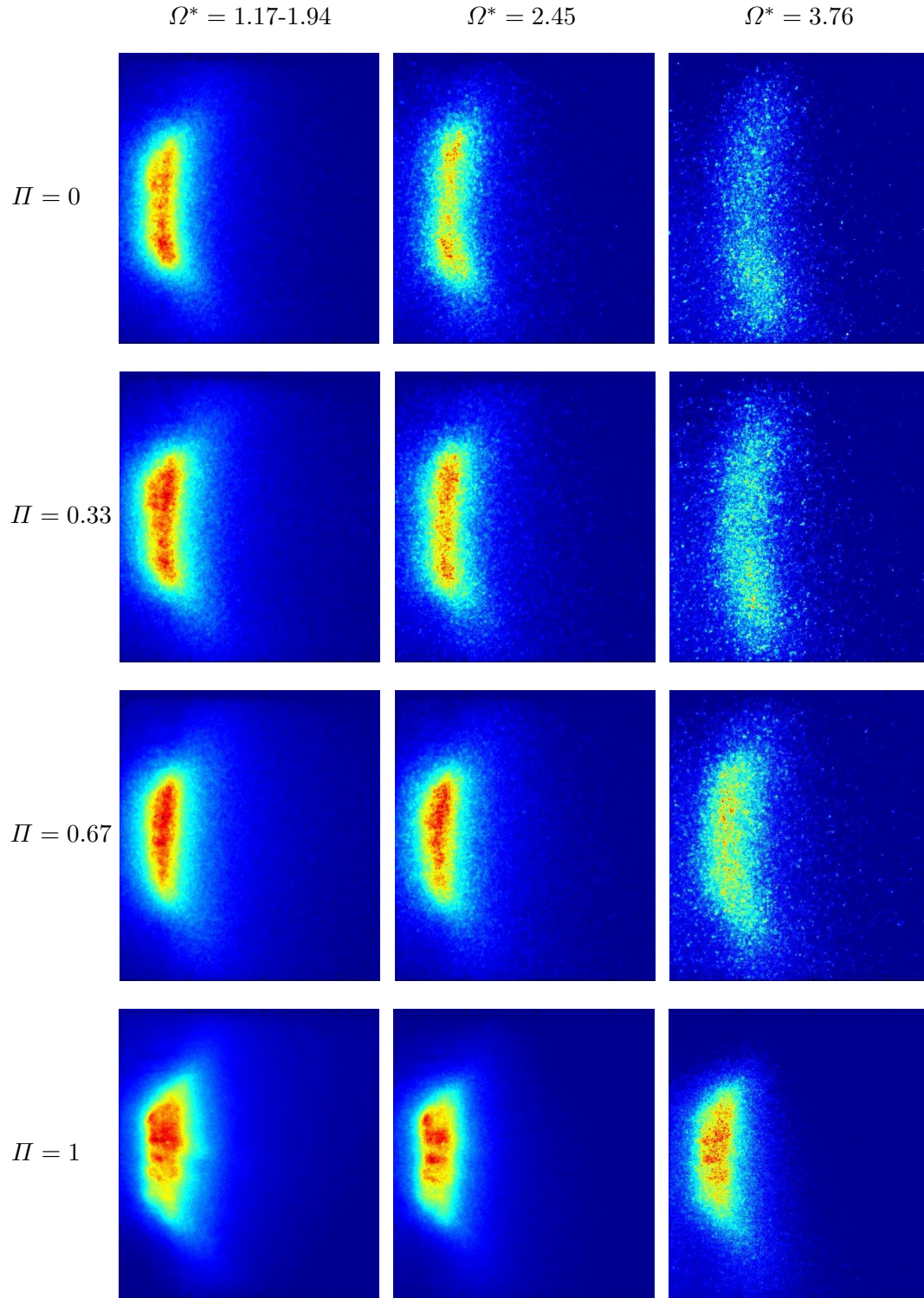


Figure 2.17: OH* chemiluminescence images; flow direction from left to right, left column: driest operating condition investigated for every O₂ injection ratio ($\Pi = 0$: $\Omega^* = 1.17$; $\Pi = 0.33$: $\Omega^* = 1.44$; $\Pi = 0.67$: $\Omega^* = 1.7$; $\Pi = 1$: $\Omega^* = 1.94$), mid column $\Omega^* = 2.45$; right column $\Omega^* = 3.76$. [162]

The flame position was calculated as the center of gravity of the OH* chemiluminescence. It is shown in Fig. 2.18 (left) to evaluate the axial movement of the flame under different

levels of steam dilution and O_2 enrichment. As stated before, the flame was shifted upstream with decreasing steam dilution and increasing flame temperature. Only the pure oxyfuel flame ($\Pi = 1$) differed from this trend as it went to a minimum distance and then moved downstream again for even lower steam dilution ratios. This effect could be attributed to the reduced momentum ratio between the steam and the reactants. Interestingly, the axial positions of all flames ($\Pi = 0-1$) collapse when being plotted over the exhaust temperature, as shown in Fig. 2.18 (left).

Figure 2.19 (left) shows that both parameters, the steam dilution, and the O_2 enrichment, strongly affect the adiabatic flame temperature. While the steam dilution decreases the adiabatic flame temperature, the O_2 enrichment has an increasing effect for a fixed steam dilution because the nitrogen mass flow, which also cools the flame, is reduced simultaneously. As the nitrogen mass flow was incorporated in the dilution ratio calculation, as done in Fig. 2.19 (right), the latter trend switched. For the fixed overall dilution ratio Ω^+ , the flames with higher O_2 enrichment revealed a lower adiabatic flame temperature because the nitrogen was replaced by steam that has a higher heat capacity and, hence, a stronger cooling effect on the flame. The exhaust temperatures are shown in Fig. 2.18 (right) over the steam dilution ratio. They followed the trends from the adiabatic flame temperature qualitatively.

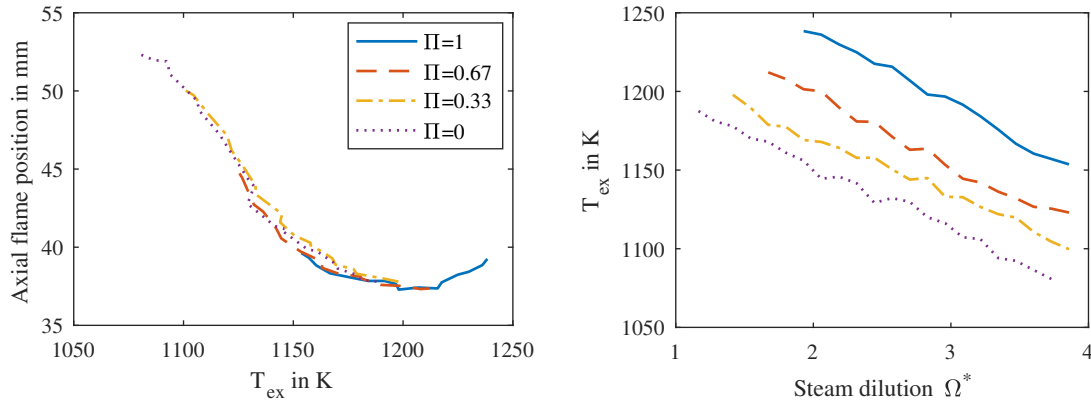


Figure 2.18: Left: Axial position of the center of gravity of the flames OH^* chemiluminescence intensity in dependence of the O_2 enrichment and the steam dilution ratio. Right: Exhaust temperatures over the steam dilution ratio. [162]

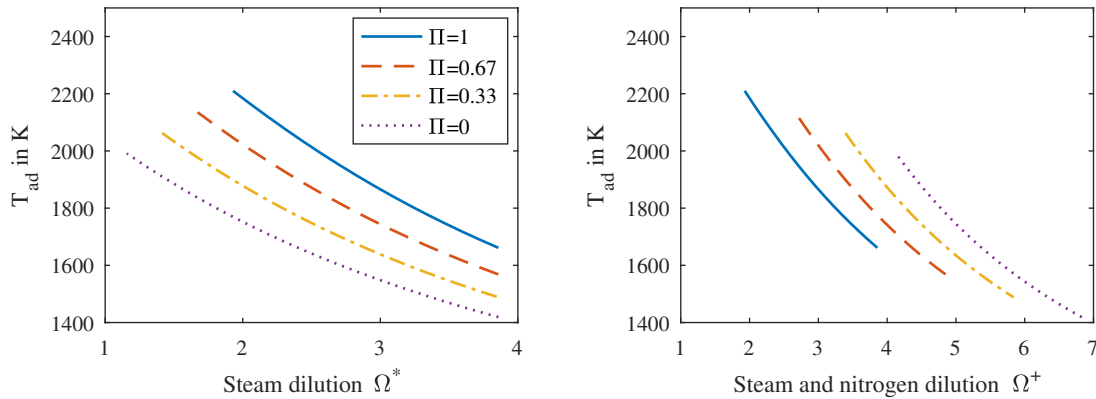


Figure 2.19: Adiabatic flame temperature over the steam dilution ratio (l.) and the overall dilution ratio (r.) for various O_2 enrichment levels. [162]

However, the gradient was much weaker. The notable difference between the adiabatic flame temperature and the exhaust temperature stemmed from the low thermal power density of the relatively large flame and from the high thermal losses of the test rig [162]. The final version of the H_2/O_2 burner featured a much smaller combustion chamber, which confined the flame in order to reach higher spatial power densities.

2.3.4 H_2/O_2 Burner: Final Version

The main investigations described in this thesis were carried out on the final version of the H_2/O_2 burner, which is depicted in Fig. 2.20. In contrast to the preliminary version, this burner used the slotted swirl generator concept of the HP burner. Additionally, the burner features a second steam passage around the O_2 injector ring and, consequently, a second independently controllable swirl generator. The swirl generators were named the inner swirler (IS) and the outer swirler (OS), respectively. They consisted of tangential slots between the plenum and two coaxial annular ducts that channeled the steam into the combustion chamber. These annular ducts were called the inner steam duct (ISD) and the outer steam duct (OSD). The swirl intensity of each swirl generator could be adjusted separately by blocking the tangential slots with the pink blocking rings depicted in Fig. 2.20. This way, two control variables were available to modify and optimize the flow field. The swirler blockage ratios of the configurations investigated in this thesis are given in Tab. 2.7 as a percentage of the total slot area. Please note that a rising configuration number corresponds

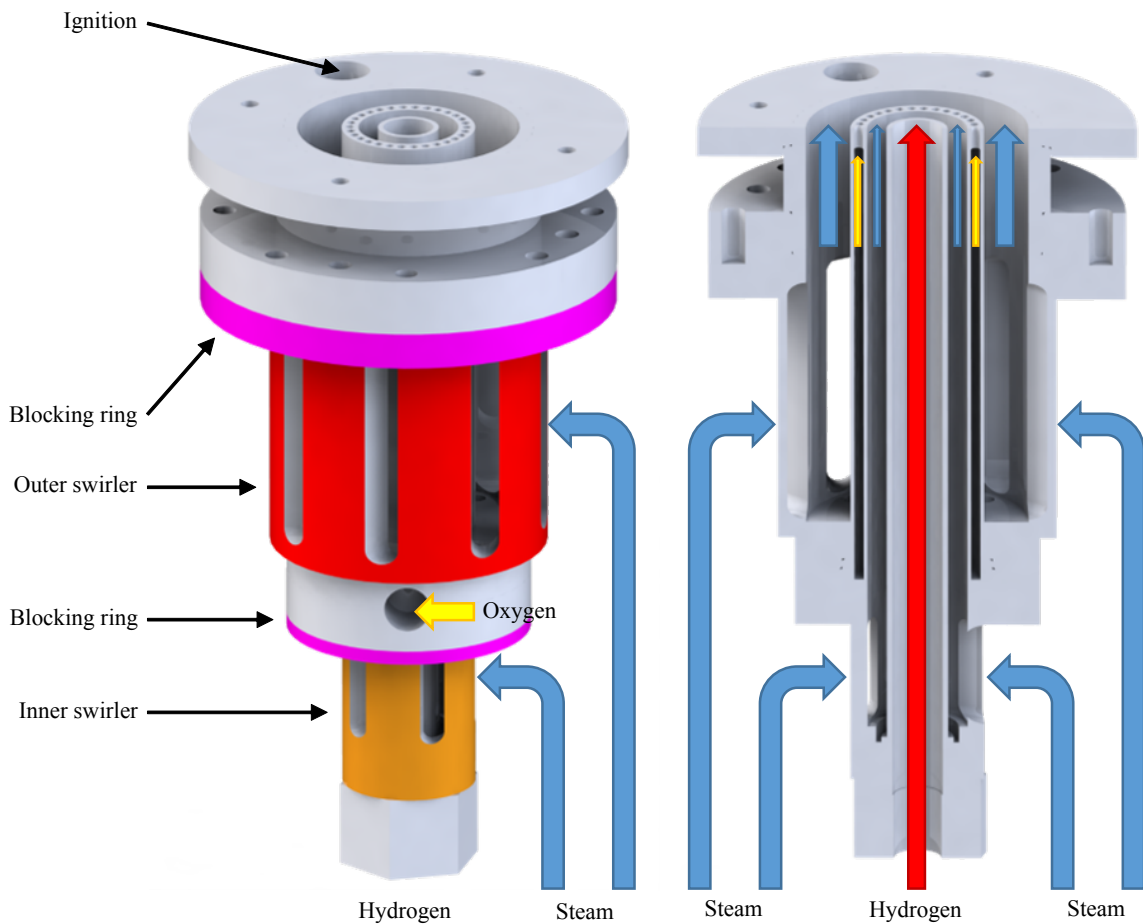


Figure 2.20: Schematic of the H_2/O_2 burner's final design.

to enhanced blockage of the swirl generator in this thesis.

Table 2.7: Combustor configurations.

Configuration	Inner swirler blockage %	Outer swirler blockage %
I	11	25
II	39	25
III	61	25
IV	83	25
V	11	50
VI	39	50
VII	61	50
VIII	83	50
IX	11	75
X	39	75
XI	61	75
XII	83	75
XIII	0	0
XIV	100	0

H₂ was injected as a non-swirling jet on the central axis, surrounded by the inner steam stream (fed through the IS). O₂ was injected through 30 coaxial holes between the ISD and the OSD. By this arrangement, H₂ and O₂ were not able to form a combustible mixture near the combustor outlet but mixed inside the combustion chamber. Thus, the risk of flashback was minimized. Additionally, the second steam duct generated an outer steam jet that protected the reactants from the cold combustor walls and minimized the latter's effect on combustion efficiency.

The combustion chamber that was attached to the burner had a diameter of $D = 80$ mm. Throughout the thesis, D is considered as the characteristic length of the combustion system. It was, therefore, used for the normalization of all length scales within this thesis. Table 2.8 displays the essential dimensions of the burner.

Table 2.8: Combustor dimensions.

Inner swirler			Outer swirler		
Number of slots	5		Number of slots	8	
Slot length	19	mm	Slot length	40	mm
Slot width	4	mm	Slot width	5	mm
Total normal area	362.8	mm ²	Total normal area	1757.1	mm ²
Inner steam duct			Outer steam duct		
Inner diameter	11	mm	Inner diameter	24	mm
Outer diameter	18	mm	Outer diameter	40	mm
Length	114	mm	Length	75	mm
Cross-section	159.4	mm ²	Cross-section	804.2	mm ²
Hydrogen injection			Oxygen injection		
Number of jets	1		Number of jets	30	
Diameter	9	mm	Diameter	1.5	mm
Total Area	63.6	mm ²	Total Area	53	mm ²

CHAPTER 3

Isothermal Burner Characterization

In the following chapter, the combustion chamber flow is characterized at isothermal conditions. The flow characteristics are of utmost importance for the combustion process itself. Naturally, the flame influences the flow field due to the sudden change in density. Nevertheless, the flow field without heat release already gives an idea of the expectable flame type and stability. Before igniting a flame in a newly designed burner, at least a qualitative knowledge of the flow field is eligible to determine specific regions in the flow, e.g., recirculation zones, velocity streaks, and shear layers. Additionally, detailed knowledge of the flow field allows for a deeper analysis of the subsequently conducted experiments in the combustion test rig.

Section 2.3.4 introduced the design of the burner, in which steam was the main flow component. In general, steam is inert and does not contain any parts of the fuel (H_2) nor the oxidizer (O_2). Hence H_2 and O_2 were injected directly into the combustion chamber of the H_2/O_2 burner. Thus, the flame can not be defined as premixed. The phenomenological distinction between a premixed and a non-premixed flame depends in this case on the relation between the progress of the mixing process and the flame position. This relation, in turn, might influence the combustion efficiency, because incomplete mixing will lead to unburned reactants in the exhaust. It is, therefore, of high interest to investigate the mixing process of the reactants.

The flow measurements in this thesis were limited to the combustion chamber. However, three-dimensional simulations allowed insights into the flow features inside the burner. Based on the simulations, the flow ratio between the swirl generators (IS and OS), the swirl number, and the burner's pressure loss could be determined.

3.1 Experimental Setup

This section describes the setup used in the experimental studies of the isothermal combustor flow. An established method to record the combustion chamber flow field is particle image velocimetry (PIV), described in Sec. 3.2.1. PIV measurements in the combustion test rig are extensive because the flow needs to be seeded with heat-resistant particles following the fluid movement. This problem is less demanding in water flow because it provides the opportunity to use particles of the same density as the fluid. Moreover, they do not stick and accumulate at the walls as oil or solid (SiO_2) seeding. The use of the highly reactive gases H_2 and O_2 in superheated steam would also imply the risk of self-ignition. For these reasons, the flow field and mixing investigations were conducted in a water tunnel (WT) test rig. Lacarelle et al. [105] conducted comparisons of combustor flows in air and water and

found good agreements for Reynolds numbers in the order of 10^4 . A comprehensive study of the similarity of the flow field, mixing, temperature distribution, and pressure loss of a can-type combustor in water, isothermal air and combustion air was reported by Clarke et al. [42]. They observed a satisfactory correlation of the velocity profiles and flow patterns quantitatively and qualitatively. They also stated that the recirculation zone generally remains similar under isothermal and reacting conditions. They were able to reproduce the concentration profiles from isothermal air flows using chemical tracers as a fuel agent in water [42].

Besides the velocity field, the mixing properties of the reactants in the combustion chamber were investigated. The planar laser-induced fluorescence (PLIF) technique was used to get a spatial distribution of the mixture. As for PIV, also the implementation of PLIF is generally less demanding in water. Instead of determining the concentration of H_2 or O_2 in steam, water was the agent for all components. Thereby the component, of which the concentration should be measured, was colored with a dye. The dye was excited by a laser and emitted light at a different wavelength. The intensity of this fluorescence was proportional to the concentration of the dye. Accordingly, the images of the dye flow facilitated the calculation of the concentration field. This method was previously applied and validated by Lacarelle [105] in the same WT test rig.

3.1.1 Water Tunnel Test Rig

The WT test rig consisted of a vertical test section with a cross-section of 400×400 mm, a 3.5 m^3 main tank, and a 0.3 m^3 tank for dyed water. It is shown as a schematic in Fig. 3.1. The main flow was driven by a radial-pump and entered the WT at the bottom. It was subsequently bent upwards and guided through a honeycomb in order to homogenize the incoming flow. The burner was placed approximately 1 m downstream of the honeycomb in the center of the test section. It was attached to a fused silica tube with the dimensions of the original combustion chamber ($D = 80$ mm, $L = 200$ mm). Downstream of this tube, the flow opened into a larger concentric volume. At the downstream end of the WT, the water drained and circulated back into the tank or a sink. The secondary and the tertiary flow were driven by two individual pumps and directly injected into the combustion chamber to simulate the reactant flows.

The main flow, coming from the large tank, was measured as volume flow by a rotating wheel flow sensor (SIGNET 8550), while a magnetic-inductive flowmeter measured the secondary flow. The tertiary flow, supplied by the dye water tank, was metered by two parallel rotameters: one for large flow rates in the range of $0\text{-}0.5 \text{ m}^3/\text{h}$ and one for higher accuracy at low flow rates in the range of $0\text{-}0.15 \text{ m}^3/\text{h}$. The locations of the sensors are depicted in Fig. 3.1. A frequency controller automatically controlled the main flow pump. If the frequency was set in the optimal range, oscillations less than 4 % of the volume flow could be achieved [105]. The secondary and tertiary flow pumps had no automated control. Instead, there was an adjustable bypass back into the tank.

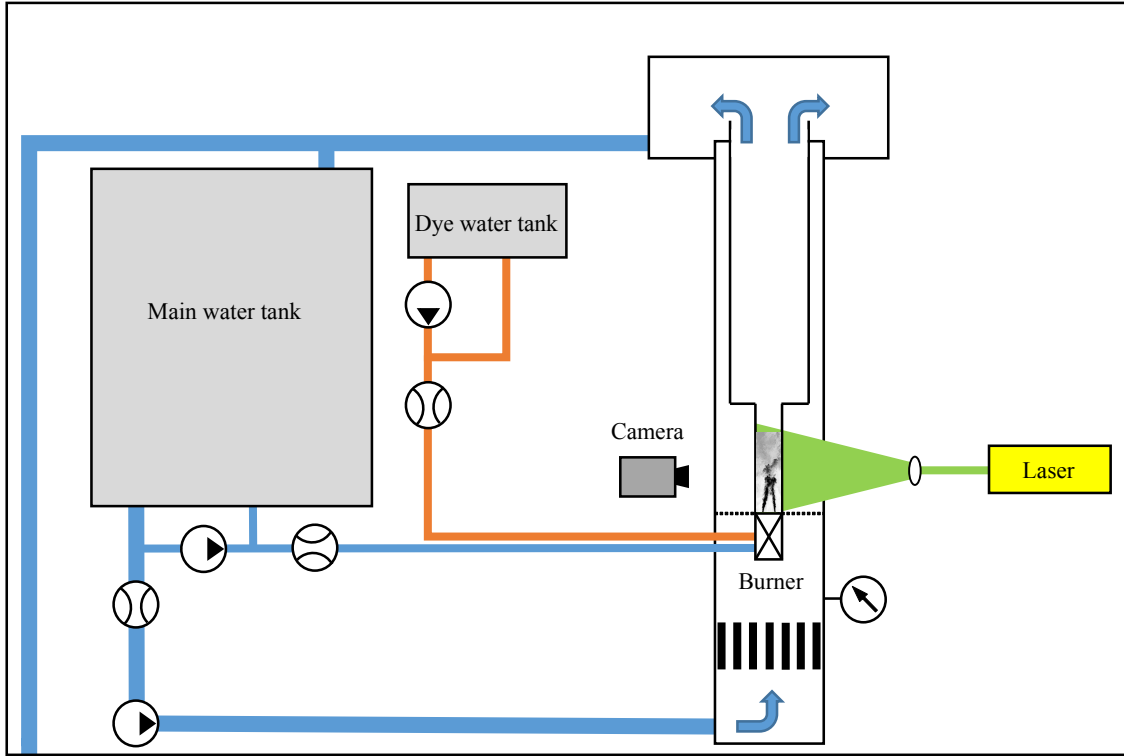


Figure 3.1: Schematic of the WT test rig.

3.2 Methods

As mentioned before, PIV and PLIF were applied for the flow field and mixing measurements, respectively. These techniques are introduced in the first two sections. Additionally, three-dimensional flow simulations have been conducted to complement the measurements and determine further flow features like the swirl number. The simulation setup is described in Sec. 3.2.4.

3.2.1 Particle Image Velocimetry

PIV is an efficient method to evaluate the flow field in a multi-dimensional manner. Velocities and flow directions are determined in the measurement plane by recording the movement of particles in the flow that scatter the light of a laser sheet. The technique and multiple extensions were described in detail by Raffel et al. [143]. In general, there are three types of PIV: planar, stereo, and tomographic PIV. The first two aim for evaluating the flow velocities in a measurement plane while the latter is applied to a volume. However, the third method is associated with high effort and requires to scan the combustion chamber from multiple angles, which is not feasible in the WT due to limited optical accessibility and refractions. The latter also accounts for stereo PIV, which makes planar PIV the most straightforward method to be used in the WT. However, planar PIV has certain disadvantages. First, the swirl-intensity can not be determined because the out-of-plane component of the flow is not measured. Additionally, a large out-of-plane component can also influence the in-plane components, which is discussed in Sec. 3.5.

The laser sheet is arranged perpendicular to the rectangular test section to avoid any refractions. A pulsed 200 mJ Nd:YAG laser with a wavelength of $\lambda = 532$ nm is used to

illuminate the particles in the flow. The seeding particles should follow the flow with a minimum of slip, be rather small and scatter the incoming laser light sufficiently. Silver-coated hollow glass spheres with a diameter of approximately $10\text{--}15\ \mu\text{m}$ and a density of $1.6\ \text{g/cm}^3$ were chosen for the study. The two-dimensional light-sheet illuminated only the particles in the center plane of the combustion chamber. The scattered light was then recorded by a 5 Hz CCD-camera with a resolution of 2048×2048 px and 14 bit. The camera axis was perpendicular to the laser sheet, and the image was scaled by a target image of the combustion chamber.

The different burner configurations were measured in two PIV campaigns (see Tab. 3.1). The spatial correlations were done in two steps with grid refinement [171]. A least-square Gauss fit at 3×3 points yielded the central peak on a sub-pixel level. Outliers were detected and interpolated by a third-order B-spline. The number of interpolated vectors did not exceed 1 %, as shown in 3.7. Additionally, median filtering with a kernel size of 3×3 was applied.

Table 3.1: PIV settings.

	Campaign I	Campaign II	
Configurations	I-IV	V-XII	
Window Size	32×32	32×32	px
Overlap	50	50	%
Scaling factor	6.445	9.41	px/mm
Pulse delay	110-200	110	μs
Flow vector field size	67×143	49×115	

3.2.2 Planar Laser-Induced Fluorescence

The PLIF measurements were conducted to study the mixing process of the main flow with the fuel and the oxidizer. The PLIF setup was similar to PIV. As previously, the steam and reactant flows were represented by water. However, the reactant flows were dyed with Rhodamine 6G for the PLIF measurements. Please note that only one reactant, either H_2 or O_2 , was dyed at a time to record both mixture fractions separately. This requires two measurements for each setpoint.

For the excitation of the dye, a continuous wave laser was employed. It had the same wavelength ($\lambda = 532\ \text{nm}$) and used the same optic path as for the PIV setup. In contrast to the PIV recordings (5 Hz), the images were recorded by a high-speed camera with a maximum frame rate of 5000 frames/s at 1024×1024 px. The Rhodamine mixture absorbed the light of the laser sheet and emitted light of a larger wavelength with a maximum at 550 nm. An optical low-pass filter was used to cut off any reflections or scattering of the laser light. As the laser intensity was not equally distributed over the measurement plane, a correction had to be applied by normalizing the images with a static intensity image I_{hom} of homogeneous dye concentration C_{hom} . Accordingly, the absolute concentration of one species C_i is calculated from the measured intensity I as

$$C_i(x, y) = C_{\text{hom}} \cdot \frac{I_i(x, y)}{I_{\text{hom}}(x, y)}, \quad (3.1)$$

$$C_{i,\text{RMS}}(x, y) = \sqrt{\left(C_i(x, y) - \overline{C_i(x, y)}\right)^2}, \quad (3.2)$$

with i being H_2 or O_2 .

Although recorded separately, both mixing processes were evaluated together as the combined mixture fraction C_{Mix} , defined in Eq. (3.4), where C_{O_2} is the mixture fraction of O_2 , i.e., the Rhodamine concentration. It is normalized by the concentration at the injection point $C_{O_2,\text{inj}}$. The same applies to the H_2 mixture fraction C_{H_2} . The mixture field $C_{\text{Mix,RMS}}$ represents the concentration's temporal fluctuations through the root mean square (RMS).

$$C_{\text{Mix}}(x, y) = \frac{\overline{C_{O_2}(x, y)}}{C_{O_2,\text{inj}}(x, y)} + \frac{\overline{C_{H_2}(x, y)}}{C_{H_2,\text{inj}}(x, y)} \quad (3.3)$$

$$C_{\text{Mix,RMS}}(x, y) = \frac{C_{O_2,\text{RMS}}(x, y)}{C_{O_2,\text{inj}}(x, y)} + \frac{C_{H_2,\text{RMS}}(x, y)}{C_{H_2,\text{inj}}(x, y)} \quad (3.4)$$

The axial development of the mixing process is quantified by the spatial and temporal unmixedness, defined in Eq. (3.6) and (3.7).

$$\sigma_0^2 = \langle \overline{C_{\text{Mix}}(x, y)} \rangle \cdot (1 - \langle \overline{C_{\text{Mix}}(x, y)} \rangle) \quad (3.5)$$

$$U_s(x) = \frac{\left\langle \left(\overline{C_{\text{Mix}}(x, y)} - \langle \overline{C_{\text{Mix}}(x, y)} \rangle \right)^2 \right\rangle}{\sigma_0^2} \quad (3.6)$$

$$U_t(x) = \frac{\overline{\left\langle \left(C_{\text{Mix}}(x, y) - \langle \overline{C_{\text{Mix}}(x, y)} \rangle \right)^2 \right\rangle}}{\sigma_0^2} \quad (3.7)$$

Here, $\overline{(\cdot)}$ denotes the temporal average and $\langle \cdot \rangle$ the spatial average in the radial direction.

In order to justify the utilization of water instead of air, Lacarelle [105] compared concentration contours in the WT and the air test rig. He qualitatively found good agreement at the burner outlet plane. Quantitatively, it must be noted that the gas flow has a much different molecular diffusivity than the dye. The effect was estimated by Reichel et al. [147] using the Batchelor scale

$$l_b = \frac{l_k}{Sc^{0.5}}, \quad (3.8)$$

which is the ratio of the Kolmogorov scale l_k and the Schmidt number Sc . It gives the scale where molecular diffusion takes place. Since Sc is in the range of unity in gaseous flows, molecular diffusion acts on the Kolmogorov scale, which is the size of the smallest turbulent eddies. In water, Sc is much higher: $Sc \approx 1000$. Accordingly, the molecular diffusion takes place on a much smaller scale than the turbulence. Assuming a constant turbulent Reynolds number, it can be concluded that the molecular diffusivity is underestimated in water. Thus, the quantitative estimation of the air-fuel mixing in the WT test rig gives a rather conservative result.

In the current burner, the reactants were injected directly into the combustion chamber by high momentum jets, creating intense shear layers (SLs). In such SLs, coherent structures arise with a much larger wavelength compared to the scales of molecular diffusion. In

addition to the stochastic turbulence, the coherent structures have a stirring effect on the mixture, which significantly dominates over the molecular diffusion [43], which in turn should be neglectable.

3.2.3 Calculation of the Volume Flows

The reacting tests in Chapter 4 and 5 were conducted at different levels of thermal power. However, to limit the experimental effort, a reference power level of $P_{\text{wt}} = 50$ kW was set for the WT experiments. Accordingly, the reactants' mass flows were fixed to the values given in Tab. 3.2, while the steam mass flow was varied to capture the influence of the level of steam dilution. In order to reproduce the correct flow patterns in the combustion chamber, the momentum ratio of the reactants to the main flow (steam) was fixed. Thus, the equivalent volume flows in the WT test rig could be computed using

$$\dot{V}_{\text{wt}} = \dot{m}_{\text{ct}} \cdot \sqrt{\frac{1}{\rho_{\text{wt}}\rho_{\text{ct}}}}, \quad (3.9)$$

where $\dot{m}_{\text{ct}}$ is the corresponding mass flow in the combustion test rig.

Table 3.2: Reactants mass flow rates in the combustion test rig and equivalent flow rates in the WT.

	H ₂	O ₂	
Molar ratio	2	1	mol
Molar weight	2	32	g/mol
Reference density ρ_{ct}	0.053	0.840	kg/m ³
Water density ρ_{wt}	998.2	998.2	kg/m ³
Water viscosity ν_{wt}	1.139e-6	1.139e-6	m ² /s
PIV Configuration I-IV			
Reference power level P_{wt}	50	50	kW
Mass flow rate $\dot{m}_{\text{ct}}$	1.5	11.9	kg/h
Water tunnel flow rate $\dot{V}_{\text{wt}}$	205	411	l/h
Momentum flux $\dot{p}_{\text{wt}}$	0.05	0.24	kgm/s ²
Reynolds number Re_{wt}	7070	2830	-
PIV Configuration V-XII			
Reference power level P_{wt}	72.5	72.5	kW
Mass flow rate $\dot{m}_{\text{ct}}$	2.2	17.3	kg/h
Water tunnel flow rate $\dot{V}_{\text{wt}}$	298	596	l/h
Momentum flux $\dot{p}_{\text{wt}}$	0.11	0.52	kgm/s ²
Reynolds number Re_{wt}	10280	4110	-
PLIF Configuration III, VII, XI			
Reference power level P_{wt}	73.0	73.0	kW
Mass flow rate $\dot{m}_{\text{ct}}$	2.2	17.4	kg/h
Water tunnel flow rate $\dot{V}_{\text{wt}}$	299	599	l/h
Momentum flux $\dot{p}_{\text{wt}}$	0.11	0.52	kgm/s ²
Reynolds number Re_{wt}	10320	4130	-

All three volume flows (steam, H₂, O₂) were calculated using Eq. (3.9). Therefore, the Reynolds numbers of the reacting tests could not be perfectly matched, but they were in the

same order of magnitude. The Reynolds number of H_2 was calculated using

$$Re_{wt,H_2} = \frac{u_{wt,H_2} D_{H_2}}{\nu_{wt}}, \quad (3.10)$$

with the bulk velocity $u_{wt,H_2} = \dot{V}_{wt,H_2}/A_{H_2}$ and the diameter of the H_2 injector (see Tab. 2.8). The same was done for Re_{wt,O_2} using the diameter of a single O_2 injector. In the case of the steam flow, the Reynolds number

$$Re_{wt,H_2O} = \frac{u_{wt,H_2O} D_{H_2O}}{\nu_{wt}} \quad (3.11)$$

was calculated from the steam injection's virtual bulk velocity and hydraulic diameter, which are defined as

$$u_{wt,H_2O} = \frac{\dot{V}_{wt,H_2O}}{A_{ISD} + A_{OSD}} \quad (3.12)$$

and

$$D_{H_2O} = \frac{2(A_{ISD} + A_{OSD})}{\pi(R_{ISD,i} + R_{ISD,o} + R_{OSD,i} + R_{OSD,o})}. \quad (3.13)$$

The steam dilution ratio was varied in the tests by adjusting the steam mass flow according to Tab. 3.4. However, the WT was limited in terms of maximum and minimum volume flows. The reference power level was increased to 72.5 kW in the second PIV campaign (Configuration V-XII) to better match the operating range of the WT with the relevant steam dilution ratios.

The last part of Tab. 3.2 shows the parameters for the PLIF measurements, which are identical to the second PIV campaign. As mentioned before, both reactants had to be measured separately. While one was dyed, the other one was uncolored. Hence, two measurements were done for each measurement point, and the supply lines needed to be switched in between. The normalization of the images required a recording of the homogeneous mixture. Since the effort of the PLIF measurements was much higher than for PIV, the number of measurements had been reduced accordingly. Table 3.3 shows the steam flow rates for the five measured steam dilution ratios.

Table 3.3: Steam mass flow rates in the combustion test rig and equivalent flow rates in the WT for the PLIF measurements.

	#	$\dot{m}_{H_2O,ct}$ [kg/h]	T_{ad} [K]	Ω [-]	$\dot{V}_{H_2O,wt}$ [l/h]	$Re_{H_2O,wt}$ [-]	$\dot{p}_{H_2O,wt}$ [kgm/s ²]
H_2	1.1	36.1	2282	1.85	1860	6210	0.28
	1.2	65.1	1845	3.34	3349	11182	0.9
	1.3	79.5	1690	4.08	4093	13666	1.34
	1.4	101.2	1515	5.19	5209	17392	2.17
	1.5	144.6	1291	7.42	7442	24848	4.43
O_2	2.1	36.1	2280	1.85	1860	6210	0.28
	2.2	65.1	1841	3.33	3349	11182	0.9
	2.3	79.5	1686	4.07	4093	13666	1.34
	2.4	101.2	1511	5.18	5209	17392	2.17
	2.5	144.6	1287	7.4	7442	24848	4.43

Table 3.4: Steam mass flow rates in the combustion test rig and equivalent flow rates in the WT for Configuration I-IV in the top table and Configuration V-XII in the bottom table.

	#	$\dot{m}_{\text{H}_2\text{O,ct}}$ [kg/h]	T_{ad} [K]	Ω [-]	$\dot{V}_{\text{H}_2\text{O,wt}}$ [l/h]	$Re_{\text{H}_2\text{O,wt}}$ [-]	$\dot{p}_{\text{H}_2\text{O,wt}}$ [kgm/s ²]
Configuration I-IV	1	38.7	1952	2.91	1991	6648	0.32
	2	44.2	1848	3.33	2275	7595	0.41
	3	49.7	1755	3.75	2557	8538	0.52
	4	55.4	1671	4.18	2851	9519	0.65
	5	60.8	1601	4.59	3129	10446	0.78
	6	66.4	1537	5.02	3419	11414	0.93
	7	71.9	1483	5.43	3700	12355	1.09
	8	77.6	1430	5.87	3993	13330	1.27
	9	83	1386	6.28	4273	14265	1.46
	10	88.3	1344	6.69	4546	15178	1.65
	11	94	1307	7.12	4840	16160	1.87
	12	99.6	1281	7.53	5124	17109	2.1
	13	105.2	1250	7.96	5413	18073	2.34
	14	110.4	1225	8.35	5685	18981	2.58
	15	115.9	1201	8.71	5967	19921	2.85
	16	121.7	1177	9.15	6263	20912	3.14
	17	127.5	1154	9.58	6564	21916	3.44
	18	132.9	1134	9.99	6840	22837	3.74
	19	138.3	1116	10.4	7119	23768	4.05
	20	143.7	1099	10.8	7398	24702	4.37
Configuration V-XII	1	36.1	2278	1.86	1859	6207	0.28
	2	43.3	2152	2.24	2228	7439	0.4
	3	50.4	2039	2.6	2596	8669	0.54
	4	57.9	1932	2.98	2982	9955	0.71
	5	65.1	1839	3.36	3352	11194	0.9
	6	72.4	1756	3.73	3726	12439	1.11
	7	79.6	1685	4.1	4095	13671	1.34
	8	86.7	1621	4.47	4461	14894	1.59
	9	94	1561	4.85	4837	16149	1.87
	10	101.2	1509	5.21	5208	17388	2.17
	11	108.4	1464	5.58	5582	18638	2.49
	12	115.5	1422	5.94	5943	19843	2.82
	13	123	1382	6.34	6331	21139	3.2
	14	130.2	1346	6.71	6703	22379	3.59
	15	137.5	1315	7.04	7077	23629	4
	16	144.5	1286	7.45	7437	24830	4.42
	17	151.7	1258	7.81	7808	26068	4.87

3.2.4 Combustor Flow Simulations

The CFD simulations in this chapter were performed using the commercial flow solver ANSYS Fluent 14.5. In order to keep the computational effort reasonable, the steady-state Reynolds-averaged Navier-Stokes (RANS) approach was chosen. Within this approach, the entire flow field is Reynolds decomposed in a mean and a fluctuating component $u = \bar{u} + u'$.

Applying this decomposition to the conservation equations followed by time-averaging leads to the RANS equations for incompressible steady-state flows:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (3.14)$$

$$\frac{\partial \bar{u}_i \bar{u}_j}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left(-\overline{u'_i u'_j} \right) + \frac{\partial}{\partial x_j} \left[\nu \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial \bar{u}_k}{\partial x_k} \right) \right]. \quad (3.15)$$

This system of equations is under-determined due to the Reynolds stresses $\overline{u'_i u'_j}$. They have to be modeled. In the current simulations, this was done using the realizable k- ε turbulence model proposed by Shih [166]. The SIMPLE algorithm was taken as pressure-velocity coupling.

The numerical setup was adapted from the simulation of the HP burner [182]. In the context of these simulations, a mesh and turbulence model study and validation had been conducted. As a result, the realizable k- ε model was found to be well suited for this kind of flow field. The tetrahedra mesh was refined at the annular steam ducts, the reactants injection, and around the entrance to the combustion chamber. Strongly skewed cells were converted to polyhedral cells in order to improve the mesh quality. Furthermore, there were five prism layers at the walls, and a wall function was used. The mesh is shown in Fig. 3.2, while Tab. 3.5 presents its statistics. The mesh in the left column was used for the parameter studies in Sec. 3.3.2. The validation of sufficient resolution was done with the finer mesh, which was scaled by a factor of 0.5 in the entire domain.

Table 3.5: Mesh statistics for the RANS simulations of Configurations I.

	RANS	RANS fine
Cells	1 773 128	6 512 634
Faces	3 908 841	14 230 897
Nodes	527 849	1 857 218
Prism layers	5	5

The combustor flow field was simulated with the WT settings, and the results were validated in Sec. 3.3.2. As in the WT measurements, water of constant temperature, density,

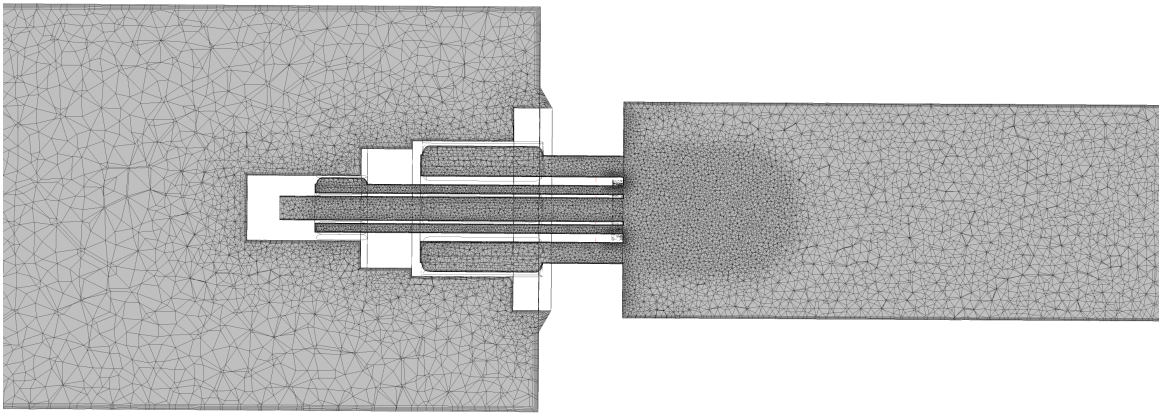


Figure 3.2: Mesh for the RANS simulations of burner Configuration I.

and viscosity represented all three flow components (steam, O₂, H₂). The velocities at the inlet boundary conditions were calculated from the volume flows in Tab. 3.2 and 3.4. Hence the Reynolds numbers and momentum ratios of the simulations matched the ones in the experiments. Since the turbulence intensity was unknown in the test rig, 5 % were assumed as inflow condition. Actually, the influence of the turbulence level at the plenum inlet was expected to be relatively low due to the significant flow acceleration in the swirler passages. The combustion chamber outlet was modeled as a vent with a loss coefficient k_L [11], describing the pressure loss over the outlet as

$$\Delta p = k_L \frac{1}{2} \rho u^2, \quad (3.16)$$

with the loss coefficient of a dump diffuser according to [82]

$$k_L = \left(1 - \frac{A_1}{A_2}\right)^2. \quad (3.17)$$

The area ratio $A_1/A_2 = 0.077$ corresponded to the dimensions of the WT. The model accounted for backflow into the domain by calculating the flow vectors from the cell layer adjacent to the outlet boundary. The near-wall flow was modeled using the enhanced wall treatment by Fluent [9]. An overview of the numerical setup including the boundary conditions (BCs) is given in Tab. 3.6.

Table 3.6: Simulation setup for the RANS simulations.

	Water tunnel domain		
Simulation type	Steady state RANS		
Turbulence model	Realizable k- ε		
Fluid	Water		
Density	Constant	998.2	kg/m ³
Viscosity	Constant	21.2	m ² /s
Temperature	Constant	293	K
BC inlet H ₂ O	Axial velocity	$f(\Omega, P_{wt})$	
BC inlet O ₂	Axial velocity	$f(P_{wt})$	
BC inlet H ₂	Axial velocity	$f(P_{wt})$	
BC turbulence	Turb. intensity	5	%
	Turb. viscosity ratio	10	
BC outlet	Vent with loss coefficient		
	Constant k_L	0.84	
BC outlet backflow	Neighbouring cells		
	Turb. intensity	5	%
	Turb. viscosity ratio	10	

In general, the complexity of the simulations was kept low to enable a large design study. Nevertheless, the simulations increase the understanding of the general flow physics in this specific combustor.

3.3 Isothermal Flow Field

Figure 3.3 shows the a typical combustion chamber flow field for a swirl-stabilized flame. Here, the combustion chamber is depicted horizontally with the flow from left to right. This

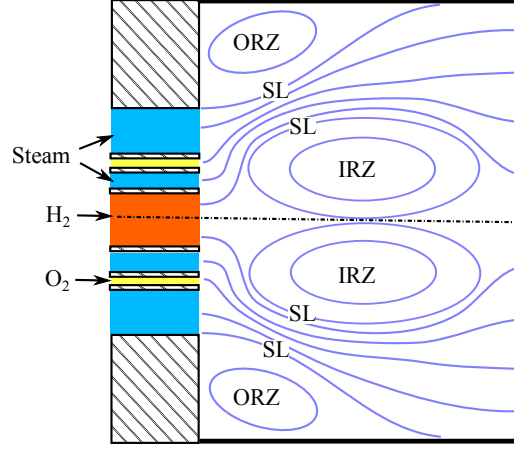


Figure 3.3: Schematic of the flow field in the combustion chamber with the shear layer (SL), the inner (IRZ) and outer (ORZ) recirculation zone.

type of representation will be kept throughout the thesis with x pointing in the bulk flow direction, and y being the radial coordinate.

The swirling motion of the steam forces the flow to spread radially and to attach to the combustion chamber walls. An outer recirculation zone (ORZ) forms in the corners between the burner and the combustion chamber wall. If the tangential momentum is high enough, the flow field undergoes vortex breakdown (VB) characterized by negative axial velocities in the center of the combustion chamber. Among many explanations for this phenomenon, Terhaar [184] described a very tangible one. According to the radial pressure equation, a vortex generates a low-pressure region on its axis. At the area jump between the burner and the combustion chamber, the flow widens up, and the radius of the vortex rises, increasing the pressure in the vortex core. Consequently, a positive pressure gradient is generated on the central axis of the combustion chamber. If the force induced by this pressure gradient is larger than the axial momentum, the flow will stagnate and eventually invert its direction. In this case, an inner recirculation zone (IRZ) forms, as indicated by the vortex ring in Fig. 3.3. In between the ORZ and the IRZ, the flow is accelerated and creates a shear layer (SL). This SL covers the whole range from negative to high positive axial velocities. This broad range includes the fluctuating flame speed. Hence premixed flames can be stabilized in the SL.

However, the existence of the VB and the IRZ depends on the swirl intensity of the flow. In the H_2/O_2 burner, only the steam flow passed the swirl generators, while the reactant jets had no swirl. Accordingly, the swirl intensity was a function of the steam dilution ratio, as shown in Fig. 3.4. Here, the swirl intensity is described by the estimated swirl number, which was calculated from the frictionless bulk flow velocities of the steam ducts and reactant nozzles (see Sec. 3.3.2). As previously mentioned, the burner featured two parallel swirl generators. The steam flow split to both according to the pressure loss of each passage. The share of the IS was quantified by the swirler splitting ratio

$$\chi = \frac{\dot{m}_{H_2O,IS}}{\dot{m}_{H_2O}}, \quad (3.18)$$

where $\dot{m}_{H_2O,IS}$ was the steam mass flow through the inner swirl generator.

Since the blockage of the swirl generators can be varied individually, each burner configuration has its χ characteristics. In Fig. 3.4 three exemplary levels of χ are shown. For

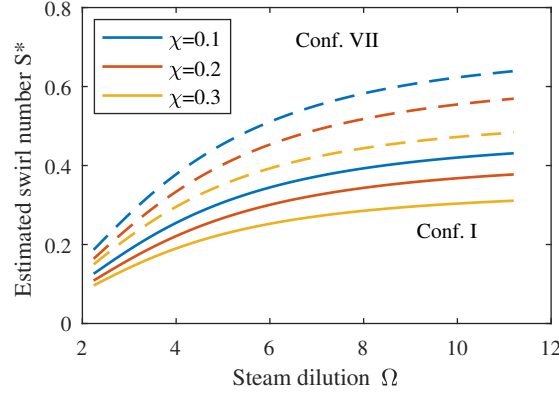


Figure 3.4: Estimated swirl number for different steam dilution ratios Ω and three levels of swirler splitting ratios χ for Configuration I (-) and VII (- -).

all three, the swirl number increases with the steam dilution ratio. However, the slope can vary for different splitting ratios, i.e., different burner configurations. Thus, for each burner configuration, it must be checked whether VB exists and at which steam dilution ratio.

The variation of the steam dilution ratio Ω in the measurements required a change of the steam mass flow. In other words, enhancing Ω increased the velocities in the combustion chamber. To still compare flow fields at different levels of steam dilution, all velocities were normalized by the bulk flow velocity at the outlet of the combustion chamber

$$u_0 = \frac{4}{\pi D^2} \frac{\dot{m}_{\text{H}_2\text{O}} + \dot{m}_{\text{H}_2} + \dot{m}_{\text{O}_2}}{\rho_{\text{WT}}} \quad (3.19)$$

3.3.1 Results of the PIV measurements

The results of the PIV measurements are divided into four parts. First, the general characteristics of the combustion chamber flow are presented with velocity and turbulence profiles. Afterwards, the discussion focuses on the IRZ and its time-depended behavior.

Please note that the flows in the following sections are named after the medium in the combustion test rig (steam, H_2 , O_2), even though the medium of all flows was water in the WT measurements.

Characteristics of the Flow Field

Figure 3.5 shows two exemplarily taken flow fields of Configuration I. The flow field is indicated by flow vectors and a contour of the axial velocity. In the left plot, a flow field with a low level of steam dilution is shown. In this case, the flow field does not exhibit the characteristics shown previously in Fig. 3.3. The tangential momentum of the swirling steam flow is low compared to the non-swirling axial momentum of the reactant jets. Thus, the swirl intensity is not high enough to cause VB. In this case, the flow field consists of the H_2 jet on the central axis and the annular O_2 jets in a weakly swirling steam environment. At $x/D \approx 0.5$, the jets merge to a single axial jet, as shown in the velocity profiles in Fig. 3.6.

Contrarily, the high-dilution flow field in Fig. 3.5 (right) shows the same flow characteristics as the schematic in Fig. 3.3. The jet from the outer steam duct is deflected outwards to the combustion chamber walls. The flow undergoes VB and forms the IRZ, which is indicated by the contour line of zero axial velocity in Fig. 3.5 (right). The main flow is accelerated around

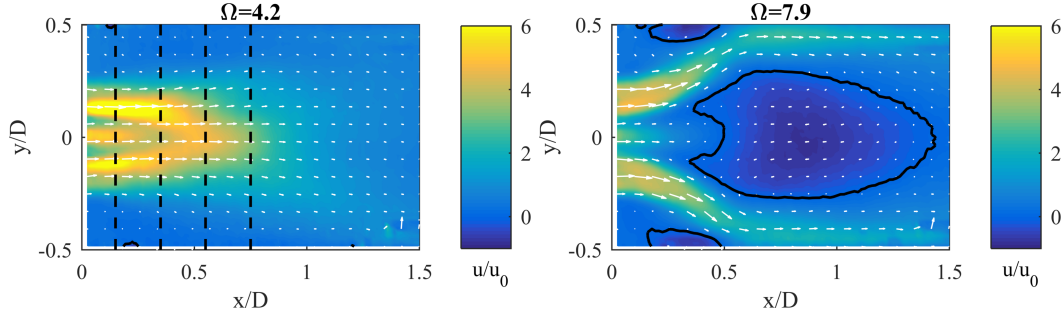


Figure 3.5: Exemplary flow field in the combustion chamber for Configuration I with low (left) and high (right) levels of steam dilution. The solid lines are contours of zero axial velocity while the dashed lines in the left plot depict the locations of the profiles in Fig. 3.6.

the recirculation bubble and provides the SLs allowing for flame stabilization. Interestingly, the H_2 jet pushes the upstream stagnation point of the IRZ downstream into the recirculation zone. It, hence, creates a concave form of the stagnation line. A similar phenomenon was previously observed by Reichel et al. [146, 148] with the H_2 -fueled HP burner. The penetration of the jet into the IRZ might influence the H_2 distribution. This issue is further discussed with the IRZ's unsteady behavior and the mixing properties in Sec. 3.4.

Flow Profiles

The flow fields of low and high steam dilution differ fundamentally. Their difference is illustrated in more detail by profiles of the axial velocity u and turbulence intensity u_{RMS} in Fig. 3.6. The profiles in the top row were measured downstream of the burner exit (0.15 D and 0.35 D) but still upstream of the IRZ in the case of VB ($\Omega = 7.9$). The bottom plots represent locations inside the IRZ (0.55 D and 0.75 D). The locations are depicted as dashed lines in Fig. 3.5 (left).

In the case of the low steam dilution (crosses), a broad jet with one central and two side peaks at $y/D = \pm 0.1$ can be observed close to the burner exit. Here, the two velocity peaks stem from the O_2 jets, and the central one represents the H_2 jet. The influence of the steam injection can be found as a little kink in the outer SL at $y/D = \pm 0.2$. There is a slight asymmetry in the profiles, which might be caused by the 30 single O_2 jets. The measurement plane was possibly not aligned equally to the jets on both sides.

At the second position (0.35 D), the central jet is mixed out by turbulent diffusion, and further downstream the two side peaks merge to a single jet. It broadens and reduces its peak velocity in the axial direction. In this sense, the flow field behaves like a round turbulent jet in the downstream part of the combustion chamber, even though there is still a tangential velocity component.

At the upstream locations, the turbulence intensity is maximal around the position of the O_2 jets. As the jet evolves through the combustion chamber, the turbulence distribution also homogenizes and forms a central maximum. Thereby, the relative turbulence intensity increases to 70 % at 0.75 D, promoting the turbulent mixing between the reactants.

The highly diluted case (circles) shows an entirely different behavior. The H_2 and O_2 momentum is the same as before, but the steam mass flow is nearly doubled. Therefore, the bulk flow velocity is higher, which decreases the normalized velocity of the reactant jets. This effect can be seen at the most upstream location, where the central peak, the H_2 jet, is much weaker than in the low-dilution case, and the O_2 jets cannot be distinguished from the steam injection.

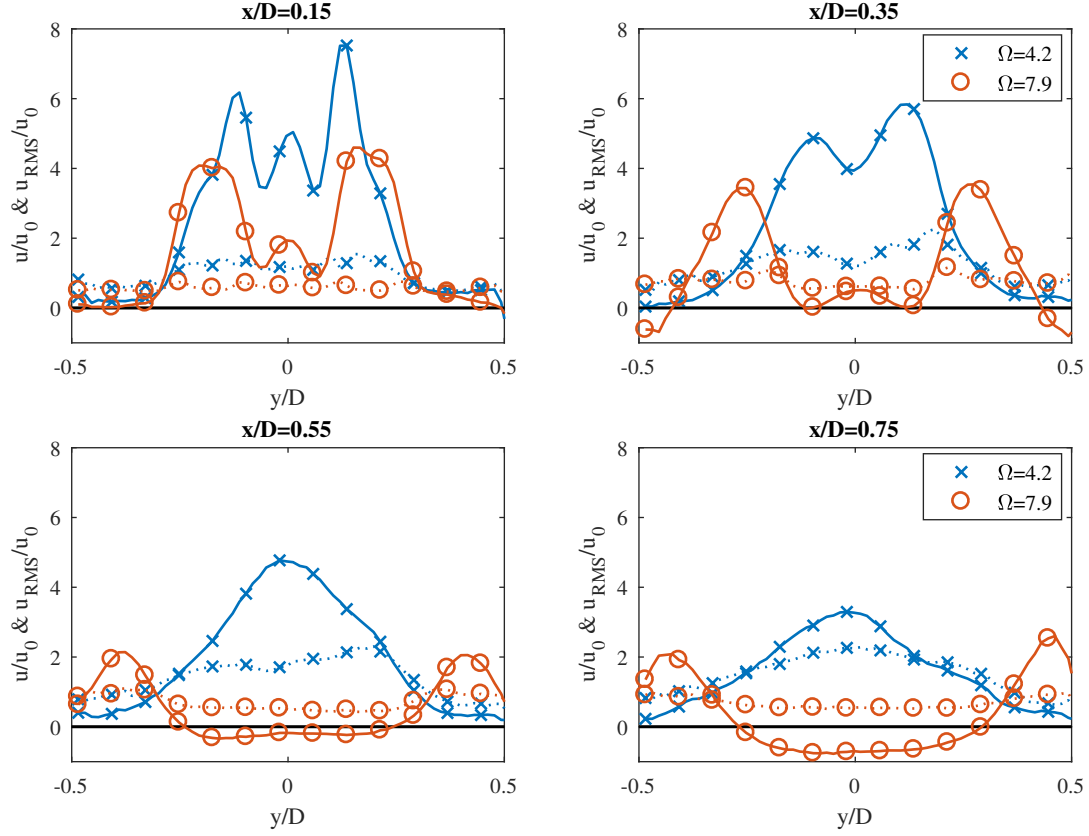


Figure 3.6: Axial velocity (-) and turbulence intensity (··) profiles at four axial positions in the combustion chamber with two levels of steam dilution.

Further downstream, the peaks are shifted radially due to the centrifugal forces of the swirling motion. At the second position ($x/D = 0.35$), the central peak is still detectable, but next to it, the axial velocity is already zero. As the flow develops further into the combustion chamber, it undergoes vortex breakdown. The corresponding negative axial velocities can be seen in the profiles within $y/D = \pm 0.25$ at the downstream locations, $x/D = 0.55$ and 0.75 . Additionally, a velocity peak evolves between the IRZ and the combustion chamber wall creating inner and outer SLs. Here, the turbulence intensity becomes maximal. Please note that even though the normalized turbulence intensities indicate less turbulence in the highly diluted case, the absolute velocity fluctuations in both cases are in the same order of magnitude.

Description of the Recirculation Zone and Design study

Swirl-stabilized combustion is associated with VB. Thus the highly diluted flow field was analyzed more in detail. In this context, a design study was conducted investigating the influence of the burner configuration on the characteristics of the IRZ.

The following four characteristic lengths depicted in Fig. 3.7 describe the shape of the IRZ: the length L_{IRZ} , the diameter D_{IRZ} , the stagnation point position x_{us} of the IRZ, and the penetration depth L_p of the H_2 jet. Thereby, x_{us} is defined as the axial position, where the axial velocity of the IRZ gets zero first. Since there is a slight asymmetry, the average of the lower and upper half-plane is taken. L_{IRZ} and L_p starts from this position. The four characteristic lengths are exemplarily evaluated for Configuration I-IV.

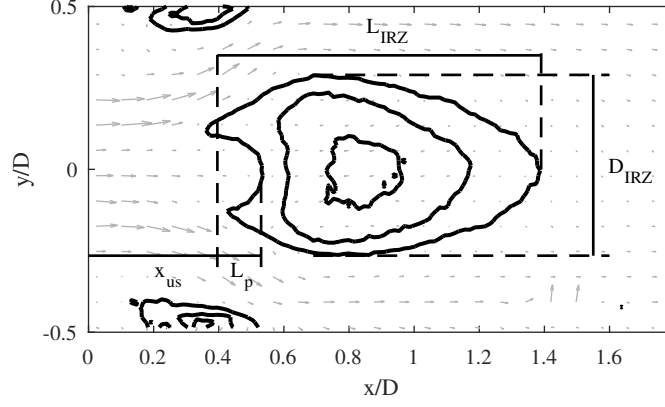


Figure 3.7: Characteristic length scales of the IRZ. The bold lines indicate the IRZ by contours of negative axial velocity.

For low steam dilution ratios $\Omega < 5$, the flow does not show VB, i.e., no IRZ exists. With increasing steam dilution, the IRZ establishes and evolves quickly in its width and length (Fig. 3.8). The rise of the width saturates at around 60 % of the combustion chamber diameter. The burner configuration has a slight influence as the higher swirl intensity increases the IRZ width. A similar trend can be found in the Configurations V-VII and IX-XII, except that the IRZ is broader in these Configurations due to the increased swirl intensity of the outer swirl generator. At Configuration XII, the IRZ width reaches 70 % of the combustion chamber diameter. The length of the IRZ shows the same characteristics as the width. It saturates between 1.1 and 1.2 D , but an explicit dependency on the burner configuration is not visible.

Figure 3.9 (left) shows the upstream position of the IRZ. At low steam dilution, the IRZ establishes at $x_{us} \approx 0.5 D$. If the steam dilution ratio is increased, the swirl intensity grows and the IRZ propagates upstream. At the Configurations I and II, this shift seems to be linearly with Ω and reaches a distance of 0.2-0.3 D for high steam dilution. In the case of Configuration III, x_{us} jumps about 0.3 D upstream, between the first and the second measurement point. However, the first point is very close to the critical swirl number, where VB occurs first. Afterwards, the movement is linear again, but the IRZ is closer to the burner exit than for Configuration I and II. This effect is related to the higher swirl intensity

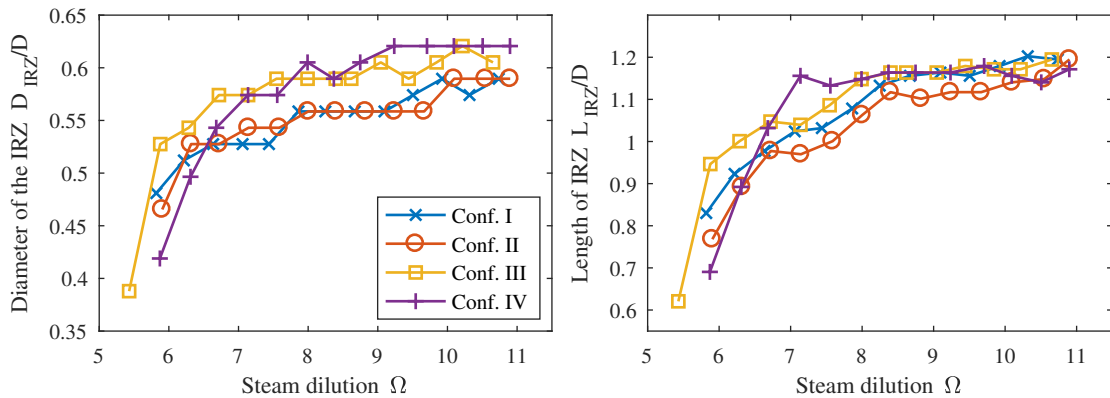


Figure 3.8: Diameter (left) and length (right) of the IRZ over the steam dilution ratio for Configuration I-IV.

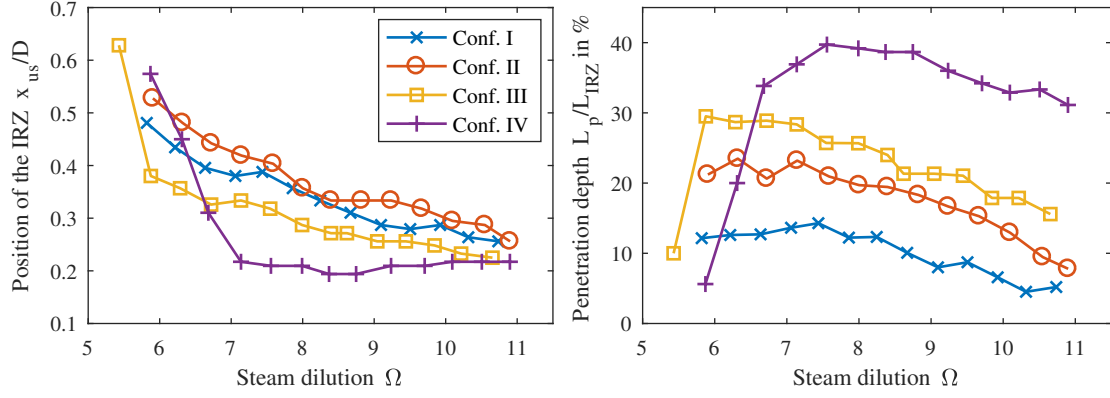


Figure 3.9: Upstream position of the IRZ (left) and penetration depth of the H_2 jet into the IRZ relative to its length (right).

due to more blockage of the inner swirl generator. At Configuration IV, the distance x_{us} is constant for $\Omega = 7-11$, and the IRZ is located very close to the burner exit ($\approx 0.2 D$). If Ω is reduced from seven, the IRZ moves rapidly downstream, even further than at the other configurations. This difference can be explained by the swirler splitting ratio χ , which will be discussed later in Sec. 3.3.2. Figure 3.15 (left) reveals that Configuration IV features a much lower χ than I-III due to the strong blockage of the IS. As a result, only 7 % of the steam flows through the ISD. As Ω is reduced, the steam mass flow rate and, hence, the tangential momentum decrease. At $\Omega < 7$, the total (inner plus outer) swirl intensity is still high enough to enable VB, but it is weak in the core flow. Thus, the stream spreads less strongly in the radial direction, and the IRZ is pushed downstream, even further than at Configuration I-III.

The best compromise seems to be Configuration III, which features a high swirl number at a reasonable swirler splitting ratio. Consequently, the position of the IRZ changes more steadily. Another measure to increase χ is to increase the swirl intensity of the outer swirl generator, which is the case at Configuration V-VIII and XI-XII, respectively. Indeed, the Configurations XI and XII show a constant distance to the burner exit of $x_{us} = 0.18 D$ over the whole range of Ω .

As previously mentioned, the H_2 jet penetrates into the recirculation zone and shifts its stagnation point downstream. The behavior of the relative penetration depth in Fig. 3.9 (right) is linked to the axial position of the IRZ. The higher the swirl intensity, the higher the penetration depth, at least for sufficient steam dilution. The reason is that, due to the higher swirl intensity, the IRZ moves upstream, allowing the H_2 jet with its constant axial momentum to penetrate further into the recirculation zone. On the other hand, the relative penetration depth of all configurations reduces slightly with increasing steam dilution because the length of the IRZ increases. The maximum penetration depth is nearly 40 % of the length of the IRZ for Configuration IV. A too deep penetration might destabilize the IRZ and should be avoided. Hence Configuration III is preferred.

Furthermore, the penetration of the H_2 jet into the recirculation zone might have an influence on the transport and mixing properties of the H_2 , which will be evaluated in Sec. 3.4.

Unsteady Behaviour

The flow field in the combustion chamber is highly turbulent, as indicated in Fig. 3.10 (left). The turbulence reaches its maximum intensity in the SLs between the IRZ and the ORZ and enhances the mixing of the reactants in this region. Inside the IRZ, the turbulence level drops significantly. To yield stable swirl-stabilized combustion, the IRZ must be stationary and no intermittent phenomenon. This requirement can be verified by analyzing the occurrences of negative axial velocities (backflow) in the flow field. Figure 3.10 (right) shows these occurrences as a probability distribution, which reaches nearly one inside the IRZ. Thus, the IRZ exists in each time instance, even though the axial velocity is fluctuating.

Further insight is gained in Fig. 3.11, which presents the complete probability density function (PDF) of the axial velocity at three locations along the centerline: upstream (A), at (B), and downstream (C) of the IRZ's stagnation point. The measured velocities are distributed in bins of 0.1 m/s, which are represented by the markers. Additionally, a normal distribution is fitted to the PDF. At the stagnation point, the velocity is normally distributed around zero axial velocity. The other two distributions show a slight skewness and do not intersect each other, which means that there is no convective transport between both points. Two crucial findings can be derived. First, due to the entirely positive axial velocity in point A, hot exhaust gas from point C cannot be transported too far upstream to the burner exit and damage components. Second, H_2 from the central injection jet cannot be carried

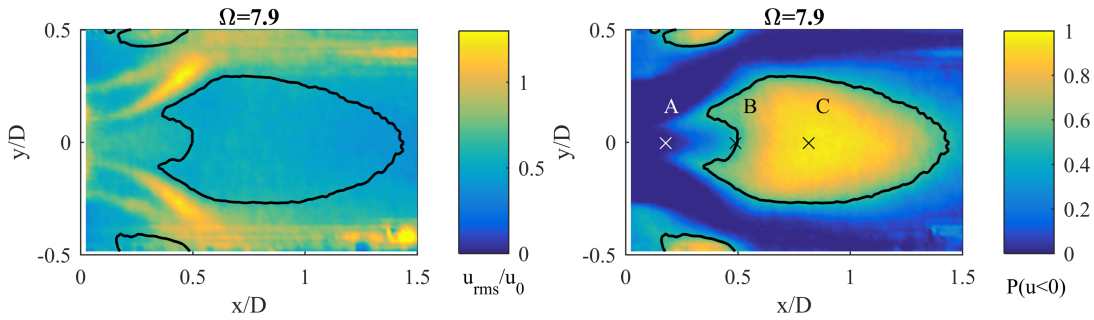


Figure 3.10: Turbulence intensity field (left) and backflow probability (right) of the highly diluted case.

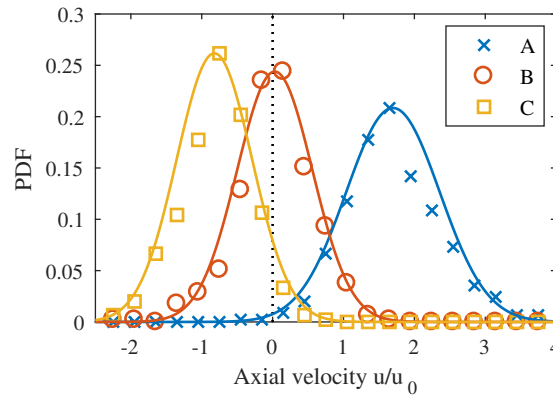


Figure 3.11: Probability density functions of the axial velocity upstream of the IRZ (x), at the stagnation point (o) and inside the IRZ (o). The three probe locations are marked in Fig. 3.10. The markers are the relative occurrences of the axial velocity at 0.1 m/s bins. The lines are fitted normal distributions.

downstream through the IRZ to the exhaust tube. Instead, it is recirculated to the SLs where it mixes with the O_2 , as shown in Sec. 3.4.

3.3.2 Results Water Tunnel CFD

There are two major advantages of the CFD simulations over the PIV measurements. First, the entire flow field of the combustor is available even regions that are not optically accessible, such as the swirl generators and steam ducts. Second, the PIV measurements of the combustion chamber yield only two-dimensional flow fields while the simulation delivers all three velocity components and, thus, enables the swirl intensity calculation.

The question to be answered by the WT CFD is, how the steam dilution ratio influences the swirl intensity. Besides, the swirler splitting ratio and the pressure loss are quantified for the different burner configurations.

First of all, the simulations are validated by comparison with the measured flow fields for Configuration I. Afterwards, the influence of the steam dilution and the burner configuration on the swirler splitting ratio and the swirl number is discussed. Based on these findings, a semi-empirical model for the swirl number is introduced. At last, the pressure loss is presented.

Validation

The flow field in the combustion chamber, calculated by CFD, is shown in Fig. 3.12 for the same steam dilution ratios, as for the PIV measurements in Fig. 3.5. In general, both cases reveal the correct flow physics, i.e., the jet behavior in the low dilution and the VB in the high dilution case. In the left plot of Fig. 3.12, the axial H_2 jet can be found on the central axis. Around this, a united jet of the steam and O_2 injections forms with very high velocities. The jets merge to one at approximately $1-1.5 D$ of the combustion chamber, which occurs more upstream in the experiments. This discrepancy indicates that the diffusion of the flow is underestimated by the simulation.

In the case of the high steam dilution ratio ($\Omega = 7.8$), the general flow physics match well. The jets from the two steam ducts are clearly visible. They stream into the combustion chamber and are immediately deflected outwards, as also seen in the experiments. While there is a weak H_2 jet on the central axis, the O_2 jets are not visible. Instead, there is a small velocity deficit between both steam ducts, because the observed plane does not cut through the O_2 nozzles. As in the PIV measurements, the flow field in the simulation forms an IRZ. However, there are two notable differences. First, the penetration of the H_2 jet into

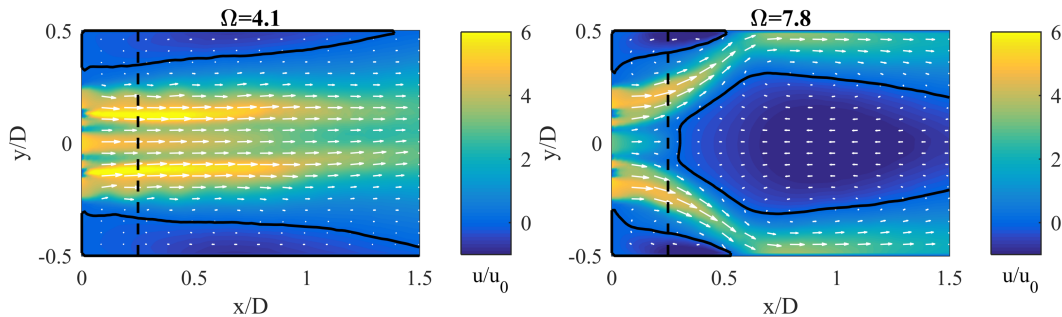


Figure 3.12: Contour of the RANS flow field in the combustion chamber for Configuration I with low (left) and high (right) levels of steam dilution. The solid lines are contours of zero axial velocity while the dashed lines depict the locations of the profiles in Fig. 3.13.

the recirculation zone is not as deep as in the measurements; second, the recirculation zone does not close but extends to the outlet of the combustion chamber. The second might stem from different boundary conditions at the outlet. While the computational domain ends at the exit of the combustion chamber, the flow undergoes a second area expansion in the experimental setup. However, these differences are not expected to significantly change the quantities of most interest: the swirler splitting ratio and the swirl intensity at the burner exit.

Next to the qualitative comparison, the simulation is quantitatively validated by velocity and turbulence profiles close to the burner exit ($x/D = 0.25$). Figure 3.13 presents these profiles for the low dilution case on the left and the high dilution case on the right side. In both cases, the velocity distributions match very well. However, there are small deviations through asymmetries. It has to be mentioned that there were measurement uncertainties regarding the volume flows in the WT, especially for the steam flow. Thus, small deviations of the absolute velocities, steam dilution ratio Ω , and the bulk flow velocity u_0 might appear. The uncertainties of the measurements are discussed in Sec. 3.5.

The turbulence profiles match perfectly in the high dilution case. In the other case, the turbulence intensity is underestimated by the CFD. The lower turbulence intensity explains the weaker turbulent diffusion, leading to a delayed merging of the jets, as observed in the paragraph before.

The independence of the grid size is validated with a finer mesh (Tab. 3.5). The fine mesh is only applied for the two cases shown in Fig. 3.13. As the profiles show no significant

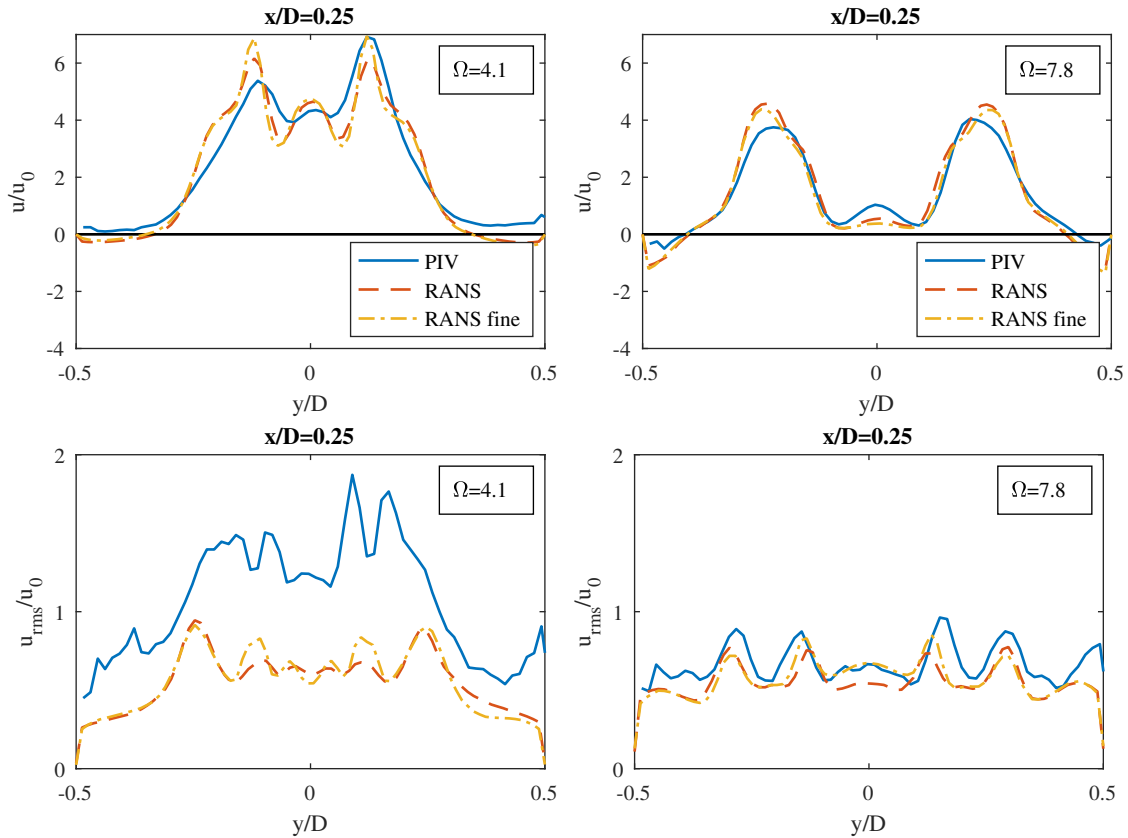


Figure 3.13: Comparison of velocity (top) and turbulence (bottom) profiles between measurement and simulation for Configuration I with low (left) and high (right) levels of steam dilution.

difference between the two meshes, the coarser one is used for the parameter studies to save computational time.

Swirler Splitting Ratio

The calculation of the swirler splitting ratio χ in Eq. (3.18) requires the steam mass flow rate through the IS. Since the mass flow is a steady-state quantity of the flow field, it can be extracted from the RANS simulations. Figure 3.14 shows that χ is around 17% for Configuration I. Thus, the mass flow through the OS is more than five times higher than through the IS. The splitting ratio decreases only slightly with the rising steam dilution ratio, which means that the change from jet to VB type flow field has nearly no effect on the flow in the swirl generators. The same holds for Configuration IV, where χ is only half of Configuration I. A variation of the thermal power is done by a proportional variation of all mass flows. Figure 3.14 shows that it does not influence on the swirler splitting ratio due to the high Reynolds numbers.

Two simulations with $\Omega = 4.2$ and $\Omega = 7.9$ are conducted for each burner configuration. The resulting χ is shown in Fig. 3.15 (left). From Configuration I to IV, the blocking of the IS slots is increased (Tab. 2.7). The velocity in the slots and the pressure loss in the IS rise, which reduces the mass flow. In other words, the mass flow is shifted to the OS, resulting in a lower χ . As the pressure loss depends quadratically on the velocity, the decline of χ gets stronger towards Configuration IV. From Configuration IV to V, there is a large jump in the swirler splitting ratio because the blocking of the inner swirl generator is set back to the value of Configuration I. Additionally, the blockage of the OS is increased from 25 % to 50 %. Therefore, the swirler splitting ratio is even higher than for Configuration I. From Configuration V to VIII, the blockage of the IS is treated in the same way as for I to IV, leading to a similar trend of χ in Fig. 3.15 (left) just on a slightly higher level. The same holds for Configuration IX-XII, where the outer blockage is around 75 %.

Since the influence of Ω can be neglected, a geometrical model can be used to describe the swirler splitting ratio. It is based on the assumption that the pressure losses for the inner and outer steam passages have to be equal: $\Delta p_{IS} = \Delta p_{OS}$. Both can be split into the pressure loss of the swirl generator and the one of the steam duct. Thereby the first is a function of the swirler blockage ratio. However, the complex flow in the swirl generator hinders a fully analytical description of the pressure loss. The following empirical model is

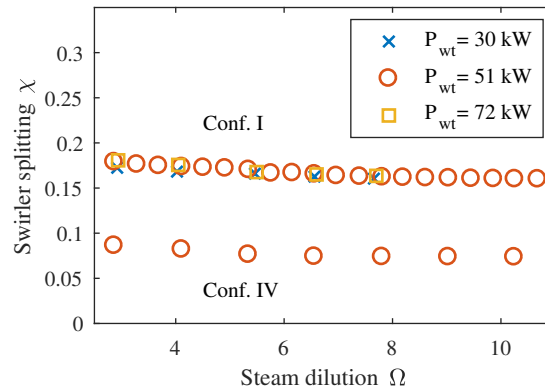


Figure 3.14: The swirler splitting ratio χ from the RANS simulations plotted over Ω for Configuration I.

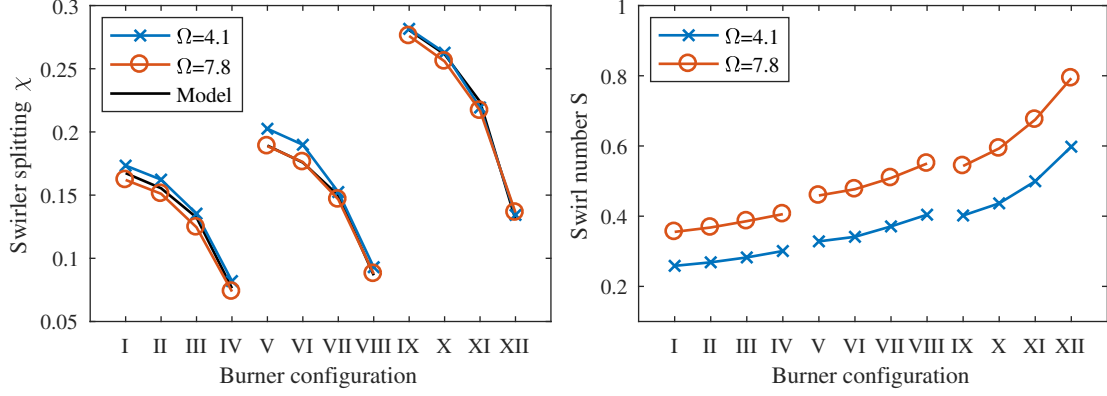


Figure 3.15: Swirler splitting ratio χ (left) and swirl number S (right) calculated from the RANS simulations for two levels of steam dilution and all burner configurations.

used for both swirl generators:

$$\zeta_{IS} = c_{IS} \left(\frac{1}{A_{IS}(1 - b_{IS})} \right)^2 + \left(\frac{L_{ISD}}{D_{ISD}} + c_{SD} \right) \left(\frac{1}{A_{ISD}} \right)^2 \quad (3.20)$$

$$\zeta_{OS} = c_{OS} \left(\frac{1}{A_{OS}(1 - b_{OS})} \right)^2 + \left(\frac{L_{OSD}}{D_{OSD}} + c_{SD} \right) \left(\frac{1}{A_{OSD}} \right)^2 \quad (3.21)$$

The dimensions are taken from Tab. 2.8. b_{IS} and b_{OS} are the blocking ratios of the swirl generators. The empirical coefficients are $c_{IS} = 1$, $c_{OS} = 0.27$ and $c_{SD} = 19$. The modeled swirler splitting ratio χ^* is calculated using the loss coefficients to

$$\chi^* = \sqrt{\frac{\zeta_{OS}}{\zeta_{IS} + \zeta_{OS}}}. \quad (3.22)$$

The knowledge of the swirler splitting ratios facilitates the calculation of the mass flows through the steam ducts and, therefore, an estimation of the swirl number.

Swirl Number

The swirl number is an indicator for the flow's swirl intensity at the entrance to the combustion chamber. It represents the ratio of tangential to axial momentum. However, different formulations can be found in the literature. In this thesis, an integral formulation will be used:

$$S = \frac{G_t}{R_{OSD,o} \cdot G_{ax}} = \frac{\int_{A_{in}} r w_{\theta} u dA}{R_{OSD,o} \cdot \int_{A_{in}} u^2 dA}, \quad (3.23)$$

where r is the distance from the burner symmetry axis, $R_{OSD,o}$ is the outer radius of the outer steam duct (Tab. 2.8), and u and w_{θ} are the axial and the tangential velocity, respectively. The integrals are evaluated on the combustion chamber inlet plane A_{in} , which is normal to the axial velocity u .

The inlet plane A_{in} includes the steam and the reactant nozzles. However, the tangential velocity w_{θ} at the reactant nozzles is zero. Hence, the reactants do not contribute to the tangential momentum, while all three components contribute to the axial one. So Eq. (3.23)

can be written as

$$S = \frac{G_{t,\text{steam}}}{R_{\text{OSD},o} (G_{\text{ax},\text{steam}} + G_{\text{ax},\text{H}_2} + G_{\text{ax},\text{O}_2})}. \quad (3.24)$$

Figure 3.16 (left) shows exemplarily the swirl number for Configuration I and VII over different levels of steam dilution Ω . In the experiments, Ω was varied by increasing the steam mass flow rate while keeping the mass flow of the reactants constant. In this case, the total tangential momentum increases while two parts of the axial momentum remain constant, as shown in Eq. (3.24). Thus the swirl number increases for higher steam dilution as it can be seen for both configurations in Fig. 3.16 (left).

On the other hand, the swirl number does not change for altered reactants mass flows (P_{wt}) as long as Ω is kept constant, indicating Reynolds-independence of the flow field. The comparison of Configuration I and VII shows a similar trend for both. At Configuration VII, the swirl number is generally higher because the tangential velocities are enhanced due to the larger blockage of the swirl generator slots.

The swirl numbers of the other configurations are evaluated and shown in Fig. 3.15 (right) for two levels of steam dilution $\Omega = 4.2$ and $\Omega = 7.9$. In both cases, the Swirl number is monotonously increasing for higher configurations, except for Configuration VIII and IX. Here, the further blockage of the OS is compensated by the release of the IS, which results in a similar swirl number for both configurations, although the splitting ratio χ varies by a factor of three. Figure 3.17 illustrates the corresponding change in the flow field. At Configuration VII ($\chi = 8.8\%$), most of the steam is flowing through the OSD, forming a broad IRZ filling about 70 % of the combustion chamber width. The recirculation zone is strongly dented by the H_2 jet. In contrast, at Configuration IX the recirculation zone is convex at the stagnation point, and the maximum velocity is reached at the O_2 jets. The IRZ is also much smaller and just fills 40 % of the combustion chamber width.

The profiles of the velocity and turbulence in Fig. 3.18, captured at $x/D = 0.25$, show significant differences between both configurations. Thereby, Configuration IX shows a much stronger recirculation in the middle of the combustion chamber and no velocity maximum on its centerline. The turbulence intensity is nearly doubled in this part compared to Configuration VIII. Moreover, in Configuration IX the turbulence intensity peaks in the SLs, which enhances the mixing of the gases and the steam.

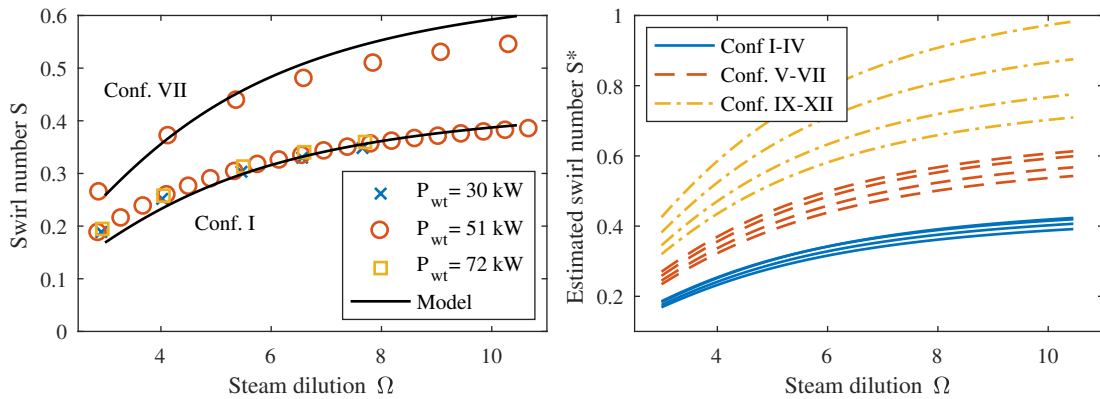


Figure 3.16: Left: The swirl number calculated by the RANS and modelled for Configuration I and VII. Right: The modelled swirl number for all Configurations. The model takes χ as input from the RANS simulations.

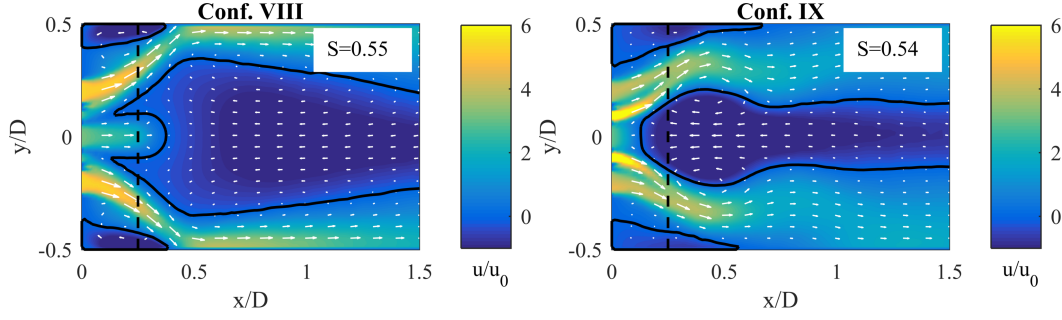


Figure 3.17: Contour of axial velocity from RANS of Configuration VIII and IX featuring the same swirl number and steam dilution ($\Omega=7.9$) but different flow fields.

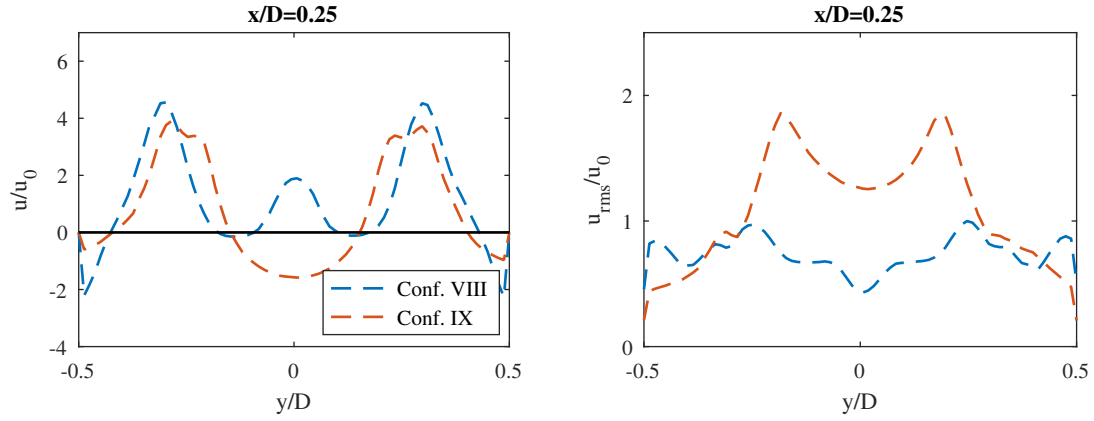


Figure 3.18: Profiles of axial velocity (left) and turbulence (right) from RANS of Configuration VIII and IX featuring the same swirl number and steam dilution ($\Omega=7.9$) but different flow fields.

Estimated Swirl Number

The swirl number, as defined in Eq. (3.23), can only be calculated from three-dimensional velocity data in the inlet plane of the combustion chamber. Since this data is not available from the PIV measurements, a flow simulation has to be done for each case to attach a swirl number to the measurement. In order to avoid this effort, an algebraic model was set up to estimate the swirl number from the geometry, the mass flows, and the swirler splitting ratio.

As stated before, the tangential momentums of the reactants is zero: $G_{t,H_2}^* = 0$ and $G_{t,O_2}^* = 0$. The axial velocities of all flows are assumed to be constant and estimated by the bulk flow velocity on each combustion chamber inlet. The bulk flow velocities are calculated from the mass flow and the area of the particular inlet $u_{0,i} = \dot{m}_i / \rho_{WT} / A_i$, where i stands either for the H_2 or O_2 injection or the steam ducts (ISD or OSD). Using this assumption, the axial momentum of the different streams can be written as

$$G_{ax,H_2}^* = u_{0,H_2}^2 A_{H_2} \quad (3.25)$$

$$G_{ax,O_2}^* = u_{0,O_2}^2 A_{O_2} \quad (3.26)$$

$$G_{\text{ax,ISD}}^* = u_{0,\text{ISD}}^2 A_{\text{ISD}} = \frac{\chi^2 \dot{m}_{\text{H}_2\text{O}}^2}{\rho_{\text{WT}}^2 A_{\text{ISD}}} \quad (3.27)$$

$$G_{\text{ax,OSD}}^* = u_{0,\text{OSD}}^2 A_{\text{OSD}} = \frac{(1-\chi)^2 \dot{m}_{\text{H}_2\text{O}}^2}{\rho_{\text{WT}}^2 A_{\text{OSD}}}. \quad (3.28)$$

In the case of the steam, the total mass flow needs to be split according to the swirler splitting ratio χ , which is given in Eq. (3.22). Please note that the density $\rho_{\text{WT}} = 998.2 \text{ kg/m}^3$ is the same for all flows. The tangential momentum is calculated from the tangential bulk velocity at the cross-section of the swirl generator slots:

$$w_{\theta,0,\text{ISD}} = \chi \dot{m}_{\text{H}_2\text{O}} / [\rho_{\text{WT}} (1 - b_{\text{IS}}) A_{\text{IS}}] \quad (3.29)$$

$$w_{\theta,0,\text{OSD}} = (1 - \chi) \dot{m}_{\text{H}_2\text{O}} / [\rho_{\text{WT}} (1 - b_{\text{OS}}) A_{\text{OS}}] \quad (3.30)$$

for the ISD and the OSD, respectively. A_{IS} is the total normal area of the inner swirler generator slots, as given in Tab. 2.8, while b_{IS} is the configuration-dependent blockage ratio from Tab. 2.7. These formulations of the bulk flow velocities yield the following momentum:

$$G_{\text{t,ISD}}^* = \frac{2}{3} \pi \left(R_{\text{ISD,o}}^3 - R_{\text{ISD,i}}^3 \right) u_{0,\text{ISD}} w_{\theta,0,\text{ISD}} \quad (3.31)$$

$$G_{\text{t,OSD}}^* = \frac{2}{3} \pi \left(R_{\text{OSD,o}}^3 - R_{\text{OSD,i}}^3 \right) u_{0,\text{OSD}} w_{\theta,0,\text{OSD}}. \quad (3.32)$$

An empirical loss term is added to account for the dissipation between the swirl generator and the combustion chamber inlet:

$$J_{\text{t,ISD}}^* = \zeta V_{\text{ISD}} \frac{1}{1 - b_{\text{IS}}} w_{\theta,0,\text{ISD}}^2 \quad (3.33)$$

$$J_{\text{t,OSD}}^* = \zeta V_{\text{OSD}} \frac{1}{1 - b_{\text{OS}}} w_{\theta,0,\text{OSD}}^2 \quad (3.34)$$

with V_{ISD} being the volume of the ISD and the loss coefficient being fitted to $\zeta = 0.0055$. Finally, the estimated swirl number reads

$$S^* = \frac{G_{\text{t,ISD}}^* - J_{\text{t,ISD}}^* + G_{\text{t,OSD}}^* - J_{\text{t,OSD}}^*}{R_{\text{OSD,o}} \left(G_{\text{ax,ISD}}^* + G_{\text{ax,OSD}}^* + G_{\text{ax,H}_2}^* + G_{\text{ax,O}_2}^* \right)}. \quad (3.35)$$

This model is validated in Fig. 3.16 (left) for Configuration I and VII. The model enables the calculation of the estimated swirl number for an arbitrary steam dilution and burner configuration, as it is done in Fig. 3.16 (right). As expected, the model shows that the swirl number rises with the steam dilution for each configuration and saturates at high dilution ratios. Moreover, the lines of Configuration IX-XII spread much more than I-IV,

which means that blocking the IS gets more efficient, the higher the OS blockage is. This observation is reasonable because the higher χ enhances the steam flow through the IS.

Pressure loss

If the H_2/O_2 burner is integrated into a power cycle, it most likely replaces external heat exchangers that usually have a high pressure loss. Nevertheless, it should be verified that a new combustion system's pressure loss is not exceptionally high. The pressure loss is calculated as

$$\Delta p = p_{\text{in}} - p_{\text{out}}, \quad (3.36)$$

where p_{in} is the pressure at the plenum inlet, and p_{out} is the area-averaged outlet pressure at the end of the combustion chamber (see Fig. 3.2).

The left side of Fig. 3.19 shows the pressure loss of Configuration I. It is given in percentage of the standard atmospheric pressure (1 atm). As expected, the pressure loss increases quadratically with the steam dilution ratio due to the increased flow velocities in the swirl generators. For the reference power level of 50 kW and the technically relevant steam dilution ratios, the pressure loss is less than 1.5 %. Please note that these values are taken from the isothermal simulations of the WT setup, using water as flow medium instead of steam, H_2 , and O_2 .

If the thermal power is increased, the pressure loss increases as well because more steam mass flow is required to keep the steam dilution ratio constant. The normalization with the dynamic pressure

$$q = \frac{\rho_{\text{wt}}}{2} u_{\text{wt},H_2O}^2 \quad (3.37)$$

makes it possible to compare the pressure loss of various burner configurations independent of Ω . This normalization makes it possible to compare the pressure loss of the various burner configurations. The reference velocity u_{wt,H_2O} is the bulk flow velocity of the steam, as defined in Eq. (3.12). The right plot of Fig. 3.19 gives this normalized pressure loss relative to the one of Configuration I.

With increasing configuration number, the swirler blockage is increased either for the IS or the OS. Thus, the tangential velocities increase and raise the pressure loss, as well as the

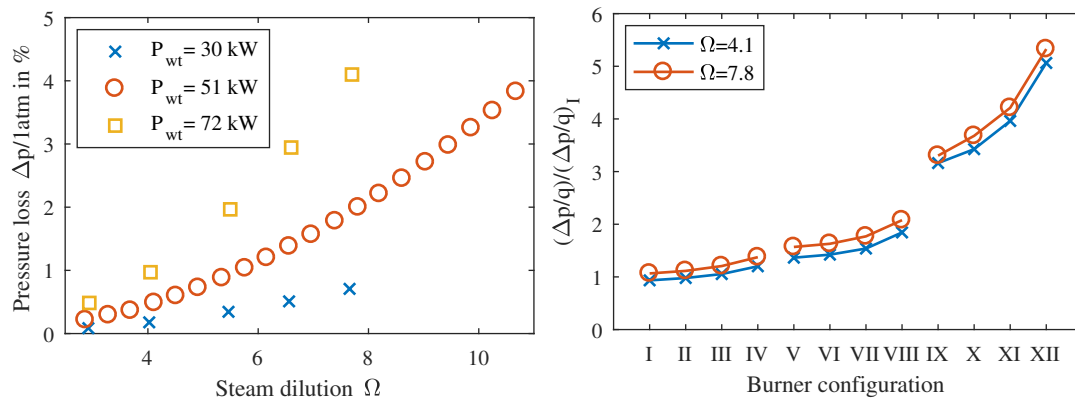


Figure 3.19: Left: The pressure loss of Configuration I normalized by the standard atmosphere and plotted over the steam dilution ratio Ω . Right: The pressure loss of all configurations normalized by the dynamic pressure and related to Configuration I.

swirl number. Configuration XII, which has the highest swirl number, reveals an over five times higher pressure loss than Configuration I. It was shown in the previous sections that the configurations VIII and IX have approximately the same swirl number. However, the pressure loss differs significantly and is about 65 % higher at Configuration IX. In general, the step from 50 % to 75 % OS blockage elevates the pressure loss drastically and should be avoided in a practical realization of the burner. Likewise, the configurations with the highest blockage of the IS (IV, VIII, XII) have a significantly higher pressure loss. Thus the following investigations focus on the Configurations III and VII. Configuration XI is added as a high-swirl design for comparison.

3.4 Mixing properties

The mixing properties of the flow were investigated for selected burner configurations. From the results of the isothermal flow field, it has been seen that a blockage ratio of approximately 60 % of the IS is favorable. Thus, the PLIF measurements have been conducted for Configuration III, VII, and XI only. As mentioned before, the measurements were done in two steps.

First, the H_2 stream was colored by the dye, while the other two were supplied with freshwater. The resulting mixture field of H_2 is shown in the top row of Fig. 3.20 for two levels of steam dilution. A single H_2 jet is streaming into the combustion chamber in both cases, but already these instantaneous snapshots reveal a very different behavior. The H_2 jet in the low-diluted case is rather compact with a low spreading rate, while in the right case, the fuel is quickly distributed radially.

In the second step, the O_2 injection was colored by the dye, as to be seen in the bottom row of Fig. 3.20. Here the difference between both cases is even more stark. In the low-diluted case, the O_2 jets penetrate straight into the combustion chamber pointing slightly inwards. The intermittency of the mixture fraction due to coherent flow structures at the jet's boundaries is clearly visible. The resulting entrainment forces the 30 single jets to spread and merge at approximately $0.5 D$ downstream of the burner exit. Here, they mix with the H_2 jet. In the case of a high Ω , the VB forces the O_2 jets to diverge from the central axis. The four instantaneous plots show that the reactant concentration in the downstream part of the combustion chamber is much more homogeneous in the high-diluted case due to the central IRZ.

An instantaneous snapshot cannot deliver quantitative and comparable information. Thus, the mixing fields must be time-averaged, which was done separately for each reactant. The time-averaged concentrations C_{H_2} and C_{O_2} are shown in Fig. 3.21, normalized by the concentration at the injection point. Afterwards, the combined mixture fraction C_{Mix} is calculated as the sum of both reactants (Eq. (3.4)) and depicted in Fig. 3.22 (top), superposed by the stagnation line of the mean flow field. The time-averaged mixing field is in good agreement with the shape of the stagnation line of the IRZ in the highly diluted case. The development of the hydrogen jet is stopped by the IRZ. The mixture cannot be convected downstream on the central axis but is guided outwards instead. The time-averaged concentration inside the IRZ is not low, but uniform and the local fluctuations $C_{Mix,RMS}$ in this region are much weaker than in the low-dilution case, as shown in Fig. 3.22 (bottom). The low fluctuations prove that fully mixed water is recirculated back from the end of the combustion chamber. In the combustion test rig, hot exhaust and possibly unburned reactants are transported back to the reaction zone by this mechanism.

The axial development of the mixing process can be quantified by the unmixedness factor

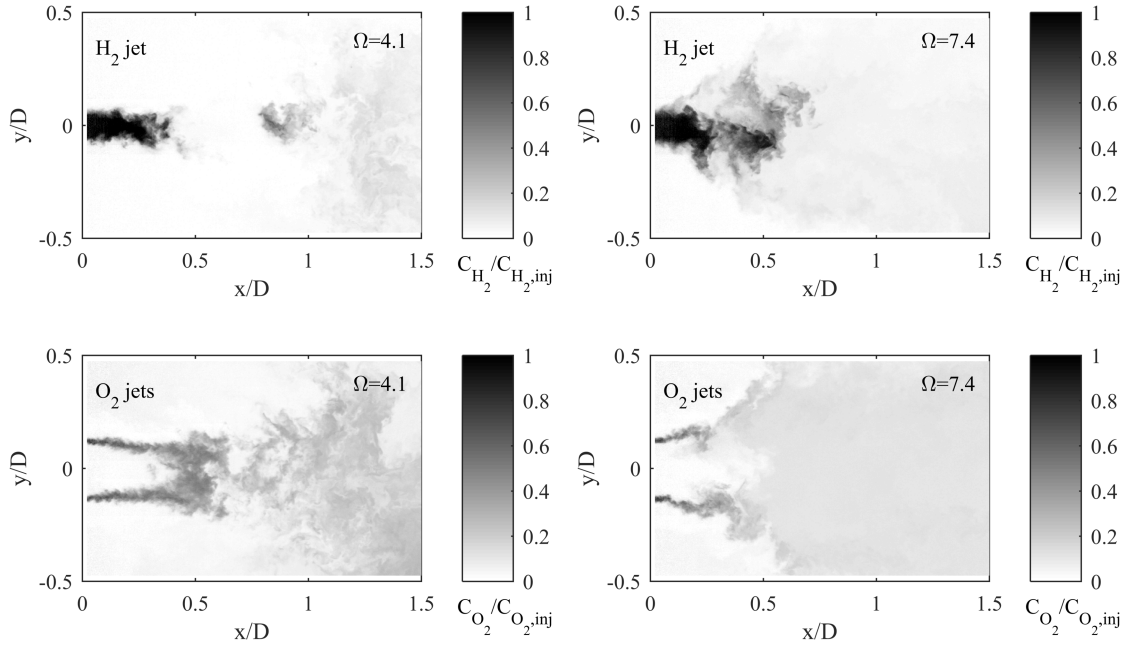


Figure 3.20: Instantaneous mixing of H_2 (top) and O_2 (bottom) in Configuration III for two levels of steam dilution.

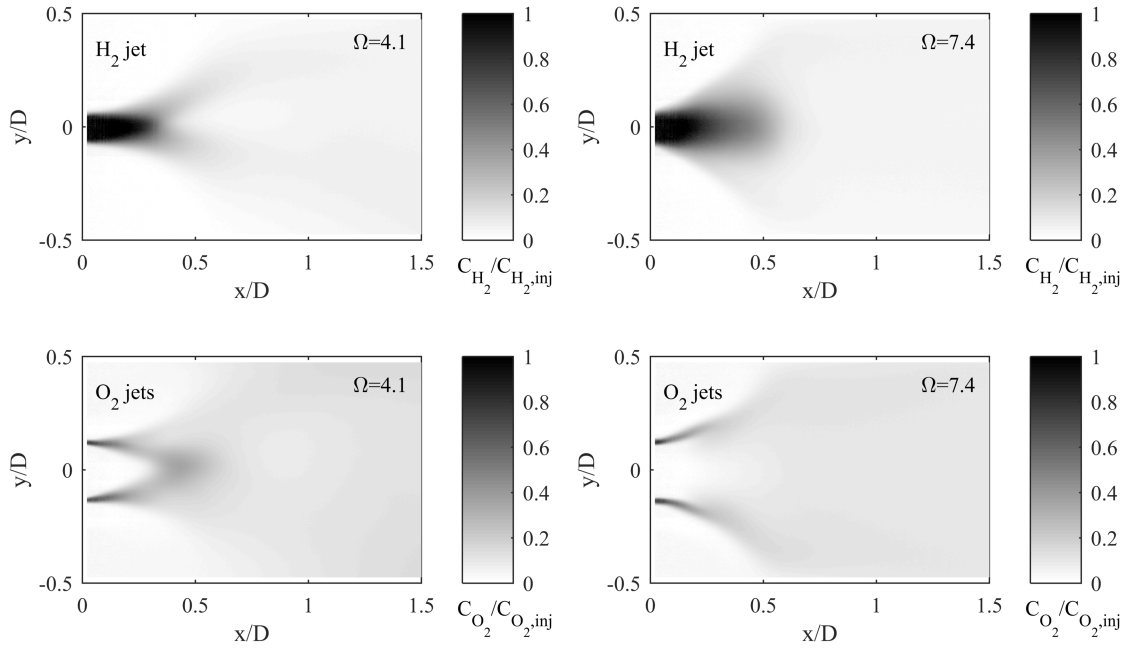


Figure 3.21: Time-averaged concentration fields C_{H_2} (top) and C_{O_2} (bottom) in Configuration III for two levels of steam dilution.

(Eq. (3.6) and (3.7)). The spatial unmixedness in Fig. 3.23 (left) is close to one at the burner exit, which corresponds to a hat-like concentration profile. Further downstream, the unmixedness drops for both dilution ratios. Thereby, the highly diluted case is continually

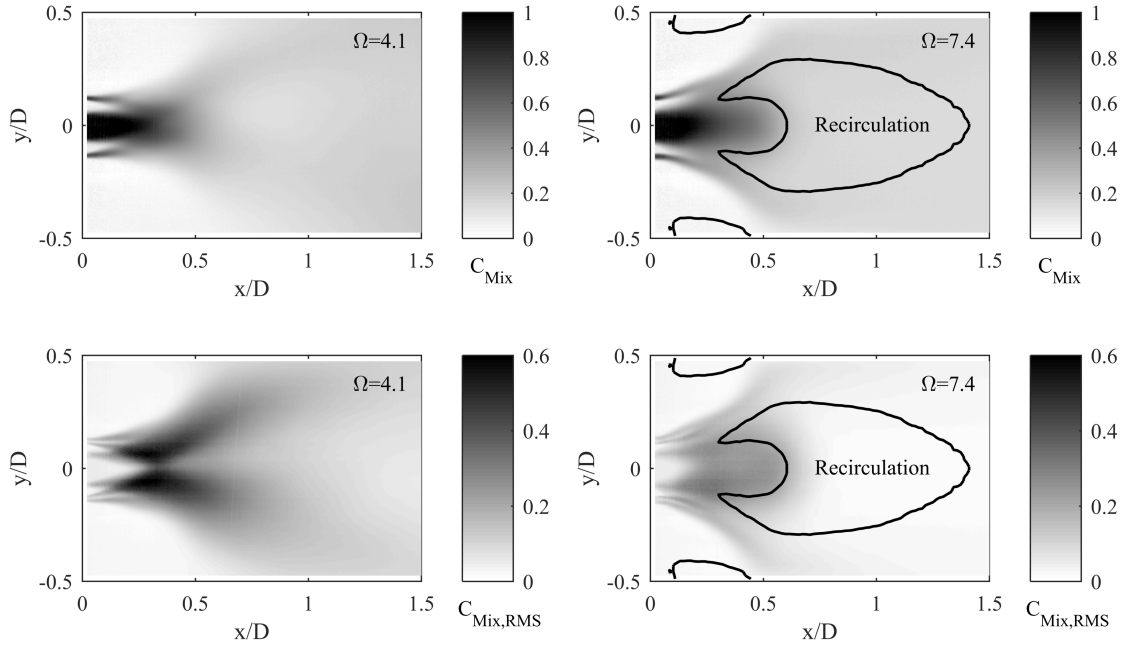


Figure 3.22: Time-averaged concentration field C_{Mix} (top) and the time-averaged fluctuations $C_{\text{Mix,RMS}}$ (bottom) in Configuration III for two levels of steam dilution.

lower, indicating a faster mixing of the reactants and the steam.

On the other hand, the temporal unmixedness of both cases grows in the first part because the jets do not start to mix with the surrounding steam immediately when they enter the combustion chamber. The induction phase becomes evident in the RMS fields in Fig. 3.22, where the unmixed flow can be seen as white cones in the jet cores. After this induction phase, a section of constant U_t follows before it starts to decline. In these two sections, the flow with high steam dilution shows a much lower value of U_t . Moreover, U_t begins to decrease earlier, and the gradient is much steeper. These findings support the statement that the mixing process is faster in the highly diluted case and that the temporal concentration fluctuations are much lower in the downstream region due to the IRZ.

As said before, Configuration XI has the highest swirl intensity of the observed configurations (III, VII, XI). Thus, the flow exhibits the largest spreading angle at the entrance to the combustion chamber, which affects the position of the mixing process, as quantified in Fig. 3.24. The spatial and the temporal unmixedness decline much further upstream in Configuration XI than in the other two. It seems that the high swirl intensity favors a fast mixing process. Configuration III and VII show a very similar behavior regarding spatial and temporal unmixedness in the highly steam-diluted case (Fig. 3.24 (right)). In the low-dilution case on the left, however, Configuration III shows faster spatial mixing between $x/D = 0.5-1$. On the other hand, it exhibits much worse temporal unmixedness, which can lead to temporal fluctuations of the equivalence ratio and, thus, unburned reactants in the exhaust stream. Hence Configuration VII should be preferred over III regarding the mixing performance. Configuration XI is superior to both.

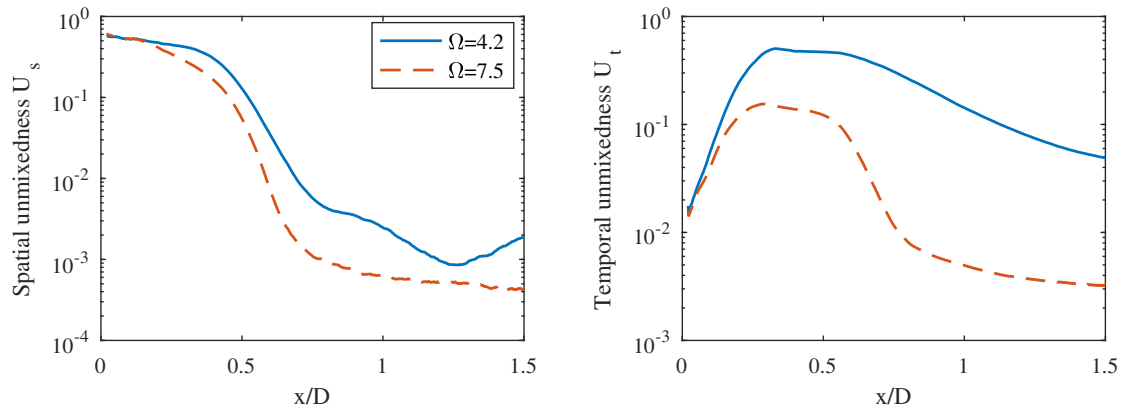


Figure 3.23: Spatial (left) and temporal (right) unmixedness in the combustion chamber of Configuration III for two levels of steam dilution.

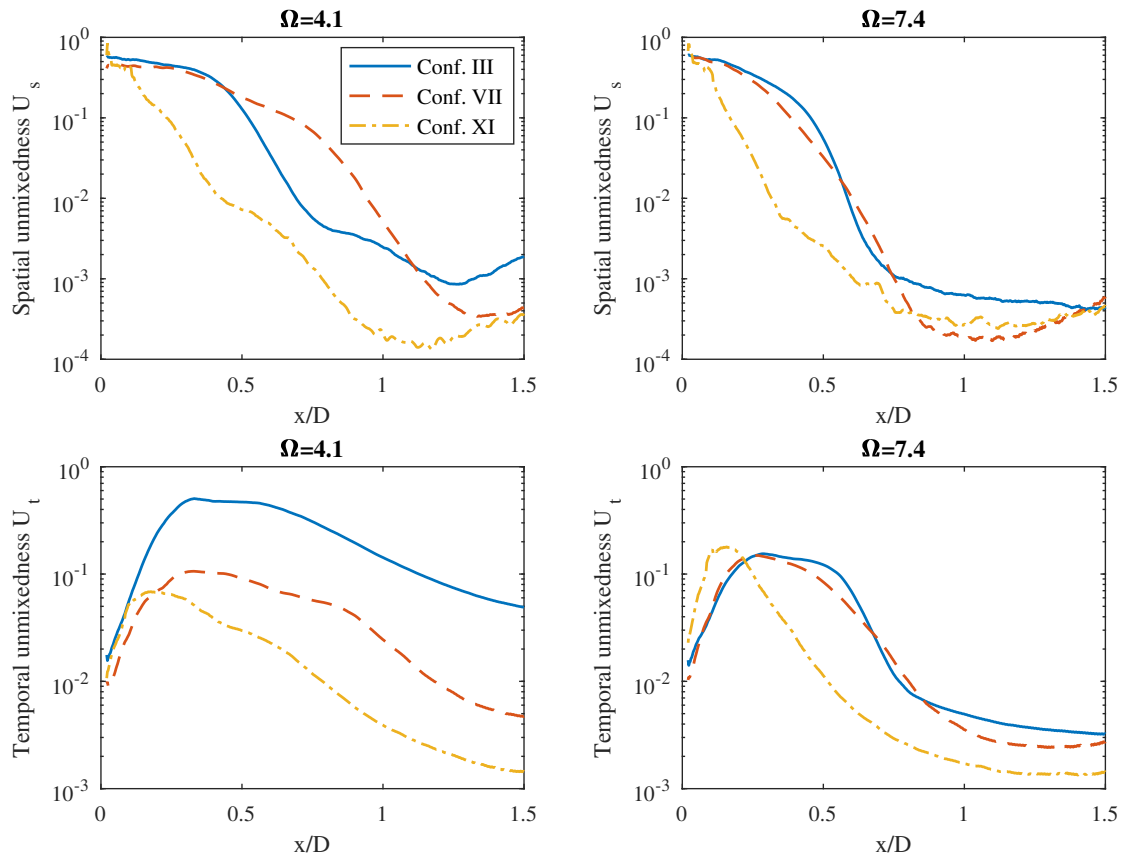


Figure 3.24: Spatial (top) and temporal (bottom) unmixedness for three configurations at low (left) and high (right) steam dilution.

3.5 Remarks on the Uncertainty of the Water Tunnel Measurements

In general, different kinds of geometrical uncertainties appear with laser-based flow diagnostics. One of these is the misalignment of the laser sheet with the measurement plane. Avoiding this error is of particular importance in the case of a swirling flow, because a misaligned laser sheet might not cut the combustion chamber in its center plane, introducing errors in the radial velocity magnitudes and the spatial velocity distribution. In the same way, the camera axis needs to be aligned perpendicular to the laser sheet to avoid image distortion. Both alignments are made more difficult due to the different refraction indexes of air (laser, camera) and water (burner) in the WT. In order to keep the optical refraction low, the round combustion chamber was surrounded by the rectangular WT shell, which contained still standing water. With this design, the refraction at the curved combustion chamber surface can be neglected because the fused silica has a similar refraction index as the water on both sides. At the outer window of the WT, the light is refracted from water to glass to air. With a refraction index of 1.333, one yields a minimum angle between the laser sheet and the camera's line of sight of 88.5° , as shown in Fig. 3.25 by the green and red lines. Thus, the resulting distortion could be neglected in the image processing.

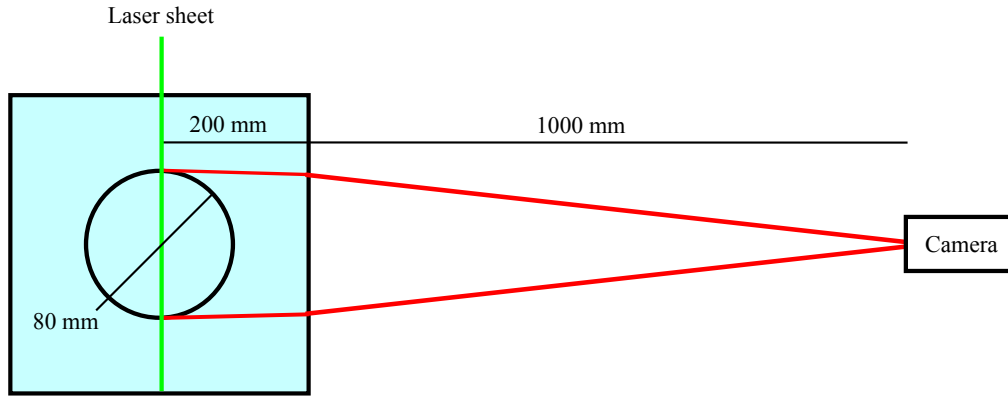


Figure 3.25: Arrangement of laser sheet and camera in the WT experiments.

However, the non-perpendicular angle still introduces a systematic error in the PIV measurements due to the out-of-plane component of the swirling flow. The perspective error was further discussed in Raffel et al. [143] and can add up to 15 % of the mean flow velocity. In the current case, the magnitude of the out-of-plane component was expected to be less than 2 m/s for Configuration XII, which was considered the worst-case because it featured the highest swirl number (Fig. 3.16) in the experiments. Taking a pulse delay of $200 \mu\text{s}$ (Tab. 3.1), the maximum pixel displacement in the out-of-plane direction would be 0.4 mm. This would cause a deviation in the radial direction of approximately 0.01 mm, respectively 0.05 m/s, which is about 2.5% of the mean flow velocity. However, Raffel et al. [143] stated that unlike random errors, the systematic influence of the perspective error does not hamper the interpretation of the flow field if the scope is to identify flow structures like in the current study.

Further error sources stem from the evaluation process of the raw images. Raffel et al. [143] estimated the overall uncertainty of the pixel displacements calculation to 0.1 pixels for the typical particle size of one pixel. With the minimum pulse delay of $110 \mu\text{s}$ and a scaling factor of 6.5 px/mm, the uncertainty converts to a magnitude of 0.14 m/s. Rukes [152] mentioned that this measurement uncertainty is normally distributed and, thus, averages out for mean quantities if the number of samples is high enough. Therefore, 451 samples for

each measurement point in the first PIV campaign and 500 in the second were evaluated. In the case of strongly nonphysical single flow vectors, these were considered to be invalid. They were filtered and interpolated (see Sec. 3.2.1). However, more than 99.3 % of the flow vectors were valid (Tab. 3.7).

Table 3.7: PIV statistics.

Configurations	Flow vectors	Valid data points	Interpolations	Other peaks
I	9581	99.8 %	0.14 %	0.09 %
II	9581	99.7 %	0.16 %	0.10 %
III	9581	99.3 %	0.46 %	0.22 %
IV	9581	99.3 %	0.22 %	0.22 %

Next to the uncertainty of the velocity field, the WT introduces uncertainties regarding the setting and measurement of the volume flows. These mainly affect the steam dilution ratio Ω . The main flow is measured by a rotating wheel flow meter (SIGNET 8550) with an uncertainty of less than 2 %. The secondary flow is determined by a magnetic-inductive sensor featuring an uncertainty of 0.5 % of the measured value. The flow from the dye tank is set using two parallel rotameters. The uncertainty here depends on the manual scale, which has increments of 10 l/h. Thus, the uncertainty ranges from 2.5 % at the lowest flow rate (200 l/h) to 0.8 % at the highest (600 l/h).

These errors also affect the dye concentration in the case of the PLIF measurements. The results of the mixing evaluation should, therefore, be regarded rather qualitatively. Due to the uncertainty of the absolute concentrations, a quantitative match of the unmixedness factors with other publications cannot be expected. Nevertheless, the mixing investigations yield an excellent insight into the mixing process, the regions of dye accumulation, and the comparison of different cases using the same measurement setup.

3.6 Summary

The PIV measurements showed that the burner generated two distinct types of flow fields in the combustion chamber: a low-swirling axial jet and a high-swirling flow undergoing VB. The type was determined by the steam dilution ratio Ω . The more steam was introduced to the flow, the higher the resulting swirl intensity in the combustion chamber, finally leading to VB. In this case, the flow exhibited a large IRZ on the central axis.

The IRZ was temporally stable and prevented species transport along the central axis of the combustion chamber, which means that the H_2 could not be convected downstream on the axis but was transported radially outwards into the SLs. There, it mixed with the O_2 stream. Furthermore, during combustion, the hot gas from the exhaust will be recirculated back in the IRZ to preheat the reaction zone. These features might improve combustion efficiency.

The CFD analysis revealed that the swirler splitting ratio χ was nearly independent of Ω , but varied with the burner configuration. As expected, the blockage of the IS decreased its mass flow to approximately 7 % at Configuration IV. Contrarily, blocking the OS raised χ up to 28 % at Configuration IX. The swirler splitting ratio directly influenced the swirl intensity, characterized by the swirl number S , which increased monotonically with the steam dilution ratio and the configuration number. A semi-empirical model was developed to enable the simulation-free estimation of χ and S . As the swirl number grew, the pressure loss also increased. Especially the Configurations IX to XII exhibited a very high pressure loss. In return, Configuration XI featured a faster and more homogeneous mixing compared

to Configuration III and VII. Chapter 5 will investigate how these factors influence the combustion efficiency.

CHAPTER 4

Burner Characterization in the Combustion Test Rig

In the previous chapter, the investigations were done in the WT under non-reacting, isothermal conditions. The next step in the burner development process was the examination of the burner's reacting characteristics.

For successful operation in an industrial application, the boundaries of the operating range must be known. The most crucial parameters of the produced steam are temperature and mass flow. They can be adjusted simultaneously by setting the thermal power and the ratio of steam dilution. These two are also the main parameters that were varied in the following investigations. To ensure stable operation over a broad range, the cold blow-off limit and the flame behavior at different steam dilution levels were determined.

Another focus was set on the analysis of the heat transfer from the flame to the combustion chamber walls. Depending on the steam dilution ratio, the temperature of H_2/O_2 can get very high and lead to a limited combustor lifetime if the walls are not adequately cooled. Therefore, the heat transfer through the wall was measured to estimate the wall temperature and the Nusselt number Nu . This information can serve as a tool for further burner developments.

The chapter is structured in the description of the combustion test rigs, the methods used and the results. The latter is further divided in three parts: the assessment of flame types and shapes, the stability range of the burner with its operational boundaries, and the results of the heat transfer measurements.

4.1 Experimental Setup

The burner was built into a combustion test rig. Two test rigs with different capabilities have been used for the combustion tests in this thesis. The main difference between combustion test rig A (CTR A) and combustion test rig B (CTR B) was the steam and reactants supply.

In CTR A, a gas-fired steam generator delivered saturated steam over a broad range of mass flows. CTR A provided larger H_2 and O_2 mass flow rates. Due to this high capacity, the rig allowed the determination of the burner's operational range in terms of thermal power and steam dilution. It was also used for the characterization of the flame shape and type.

CTR B featured a smaller electric steam generator, limiting the steam mass flow to approximately 60 kg/h. In return, the steam could be superheated and delivered to the burner's plenum at temperatures up to 650 K. The preheating significantly improved the ignitability and represented a more realistic boundary condition regarding industrial processes (see Chapter 2). Matching the high-temperature inlet conditions is necessary for heat transfer and combustion efficiency measurements to deliver relevant and comparable results. Hence, CTR B was used for the investigations in Sec. 4.5 and Chapter 5. The general capabilities

of both test rigs are given in Tab. 4.1.

Table 4.1: Comparison of the capabilities of CTR A and CTR B.

Combustion Test Rig	A	B	
Steam mass flow rate	20-140	20-60	kg/h
Plenum temperature	375	375-600	K
Thermal power level	0-70	0-35	kW

4.1.1 Combustion Test Rig A

A schematic setup of CTR A is shown in Fig. 4.1. A gas-fired steam generator delivered saturated steam at 375 K. The reactants, H_2 and O_2 , were directly supplied to the burner flange from high-pressure (300 bar) gas bottle bundles. The steam-filled plenum preheats them. See section 2.3.4 for a detailed burner description and Appendix A for a simulation of the flow in the fuel lines. The burner was mounted into a plenum with a diameter of 170 mm and a length of 220 mm. An optical accessible quartz glass combustion chamber with a diameter of $D = 80$ mm and a length of $L_{cc} = 200$ mm was attached to the burner outlet. Downstream of the combustion chamber, the flow was expanded to two 105 mm diameter water-cooled exhaust tubes with a total length of 810 mm. The exhaust tube released the hot combustion gases through a 40 mm orifice to the atmospheric extraction unit. Two Coriolis flowmeters (*E+H Proline Promass 80A*) measured the reactant mass flows with an uncertainty of 0.5 % of the value. The valves were set to the stoichiometric ratio of H_2 and O_2 . A further Coriolis flowmeter measured the steam mass flow, which was kept constant by a proportional integral derivative controller (PID).

In the measurements, the reactant flow rates were fixed to a specific thermal power level, while the steam mass flow was increased from the minimal possible amount (20 kg/h) to blow-off (between 50 and 140 kg/h) in order to vary the steam dilution ratio Ω . The H_2/O_2 flame was ignited by a small natural gas/air flame using a spark ignition box (*Hegwein ZTF 24*). As soon as the H_2/O_2 flame burned stably, the ignition flame was turned off.

The exhaust temperature was measured by an unshielded thermocouple inside the exhaust tube, 260 mm downstream of the combustion chamber inlet (see Fig. 4.1).

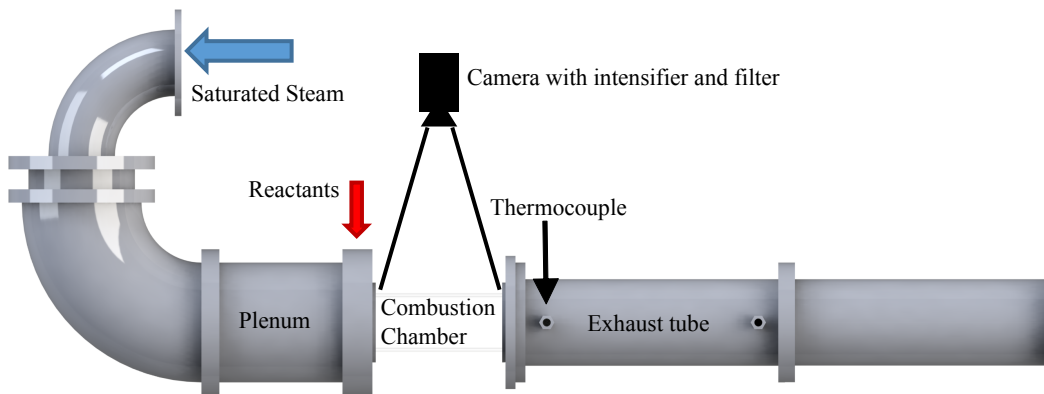


Figure 4.1: Schematic setup of the combustion test rig A.

4.1.2 Combustion Test Rig B

The main difference between CTR A and CTR B was the steam supply capability, as stated in Tab. 4.1. The steam mass flow rate was measured by a swirl flowmeter with an uncertainty of 0.5 % of the measurement value and set by a PID controller. In contrast to CTR A, the O₂ line was split into a main and a secondary line to allow for automatic control of the equivalence ratio. The control mechanism is described later in Sec. 5.2.1. The current section explains the experimental setup for the heat transfer measurements.

In CTR B, a stainless steel (Type 1.4571) tube replaced the fused silica combustion chamber to evaluate the heat transfer from the combustion products to the wall. A co-flow of air actively cooled the combustion chamber, which was necessary to keep the wall temperature in the non-critical range of the material. The cooling air flowed through four inlets into an annular duct around the combustion chamber, as shown in Fig. 4.2. The air was provided from a pressurized tank, and its mass flow was measured and kept constant by a PID controller. A thermocouple in one of the inlets monitored the inlet temperature. The air entered the annular duct approximately 15 mm downstream of the combustor head. Two slotted sheet metals at the beginning and the end of the annulus distributed the air equally and enhanced the turbulence intensity in the duct. Eight measurement points with Pt-100 temperature sensors with an uncertainty of $0.15^{\circ}\text{C} + 0.2\%$ were distributed along the length of the annulus at a distance of 20 mm from each other. The sensor heads were placed in the center of the annulus, about 8 mm away from the outer combustion chamber wall to ensure being outside of the thermal boundary layer. At three axial position, additional sensors were installed around the circumference. The cooling chamber had two diagonally arranged outlets, which exhausted the cooling air into the ambient. The outlets featured a thermocouple each to record the outlet conditions. The entire combustion chamber was thermally insulated to reduce heat losses other than due to the active cooling.

Table 4.2 gives the setup's dimensions. Compared to the fused silica one, the metal chamber was 10 % longer, but the actively cooled section had the same length.

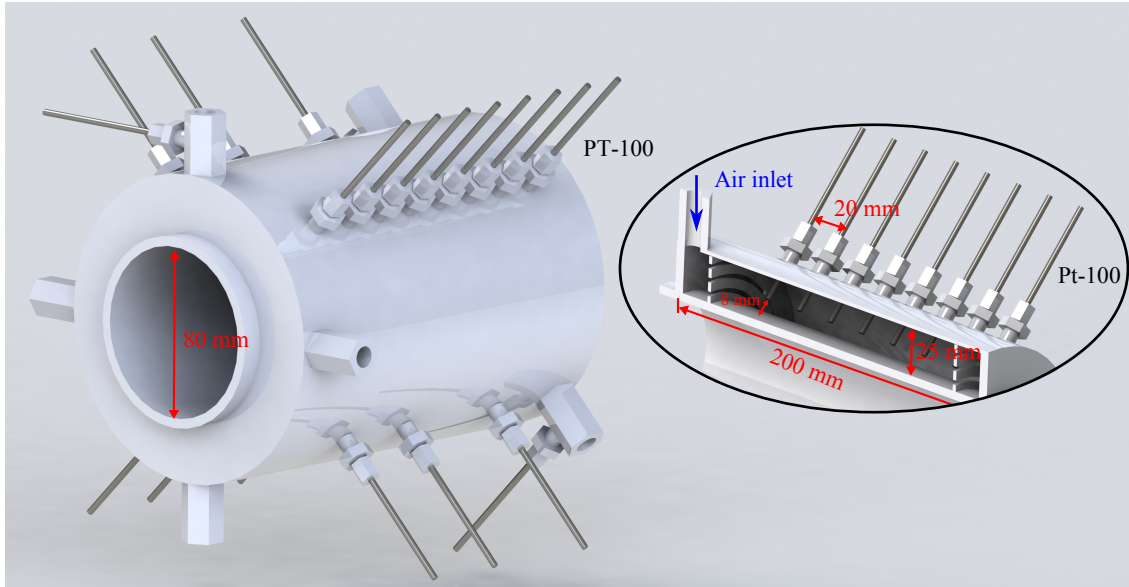


Figure 4.2: Rendering of the steel combustion chamber, used for the heat transfer measurements. The left end of the metal tube was mounted directly to the burner head.

Table 4.2: Dimensions of the fused silica and the metal combustion chamber.

Combustion Chamber		Fused silica	Metal	
Length combustion chamber	L_{cc}	200	220	mm
Length cooling section	L_{duct}	200	200	mm
Diameter combustion chamber	D	80	80	mm
Wall thickness	s_w	5.2	4.5	mm
Cooling mechanism		ambient air	air co-flow	
Cooling mass flow	$\dot{m}_{air}$	-	25	kg/h
Inner diameter cooling duct	$d_{duct,i}$	-	89	mm
Outer diameter cooling duct	$d_{duct,o}$	-	140	mm
Hydraulic diameter cooling duct	$d_{duct,h}$	-	51	mm

4.2 Methods

This chapter focuses on the results of two distinct measurement campaigns: recording flame types and operational maps, and measuring heat loss. The measurement techniques for both are introduced separately in the following subsections.

4.2.1 OH* Chemiluminescence

A common method to evaluate the location and the heat release of a flame is to record the chemiluminescence emission of species in the reaction zone. In the case of hydrocarbon flames, the hydroxyl radical (OH*) and methylidyne radical (CH*) are typically used for this purpose. In the current study however, the burner is operated on pure H₂, so no CH* is present in the combustion process. There is a wide-spread idea of H₂ flames being invisible to the human eye, which is not true. To clear this myth, Schefer et al. [158] conducted measurements of the emission spectrum of H₂ flames. As shown in Fig. 4.3, even in the visible part of the spectrum (400 to 780 nm), there is a weak emission called blue continuum, next to the light emission of heated steam, which is the product of the reaction. However, the intensity at these wavelengths (> 350 nm) was scaled by a factor of 6.5 in the figure to increase the perceptibility. Hence, the much more prominent emission is the sharp peak at approximately 308 nm in the invisible range, which can be attributed to the chemiluminescence of OH*.

OH* is an excited radical, which arises in the reaction and is generally unstable. When it returns to its electronically ground state, it might emit a photon. This process takes place in the order of 700 ns [113], which corresponds to a negligible displacement in the flow. It can, therefore, be used to detect the reaction zone in hydrocarbon flames. Moreover, the integral heat release rate can be derived from the integral OH* intensity, which has been shown in various studies, e.g., Haber et al. [76]. However, the proportionality of the local heat release distribution to the OH* intensity is only valid for premixed laminar flames. Lauer et al. [113] summarized that the chemiluminescence signal is affected by the turbulent intensity, the strain rate, the curvature, the degree of premixing, and others factors that are not temporally or spatially constant. In the investigated burner, especially the degree of premixing is unknown and varies much for the different flame types. For this reason, the OH* chemiluminescence analysis focuses more on the localization of the flame than on the rate of heat release.

The light emission was recorded by a high-speed camera (*Photron FASTCAM SA 1.1*) with a frame rate of 125 Hz. A total number of 3640 images have been acquired per data point.

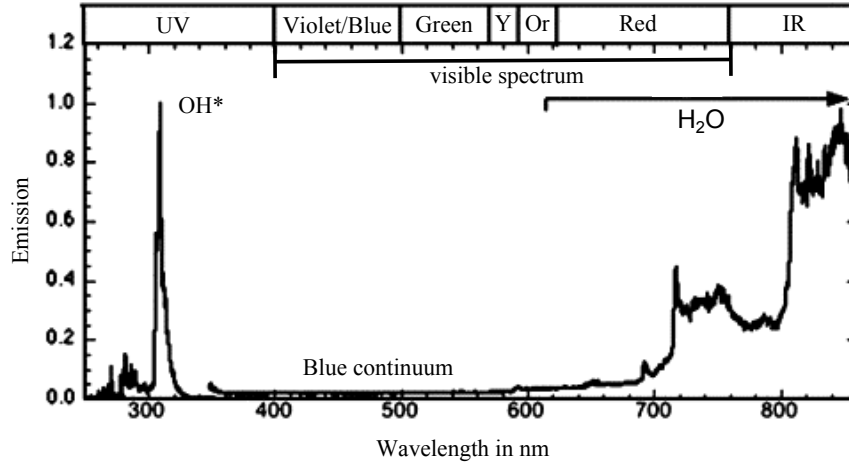


Figure 4.3: Normalized emission spectrum of a premixed H_2 -air jet flame, taken from Schefer et al. [158]. Please note that the emission intensity between 350 and 850 nm was scaled by a factor of 6.5 to increase the perceptibility.

The incoming light was filtered to capture the OH^* chemiluminescence solely. Figure 4.4 shows a close up of the OH^* peak in the emission spectrum of an H_2 /air flame, taken from Zhao et al. [205]. The peak is centered at 308 nm and has a width of approximately 20 nm. Hence, the UV transparent camera optic was equipped with a band-pass filter (320/40 nm, 60 % transmission). Its cut-off frequencies are shown as dashed lines in Fig. 4.4. The weak light from the chemiluminescence was intensified by an electronic image intensifier (*HiCatt, Lambert*), mounted in between the camera and the band-pass filter. In the post-processing, the images were averaged and deconvoluted by an inverse Radon transformation [94].

In accordance to the investigations in Chapter 3, the flames were recorded for Configuration III and VII. Since the different burner configurations were quite similar, no further evaluation was necessary. The most interesting parameters in the reacting test were the thermal power level P_{th} , the steam dilution ratio Ω , and the adiabatic flame temperature T_{ad} , which are given in Tab. 4.3. Additionally, the Reynolds numbers are given for comparison with the WT measurements. The thermal power level was fixed to 50 kW. The corresponding reactant flow rates can be found Tab. 4.4.

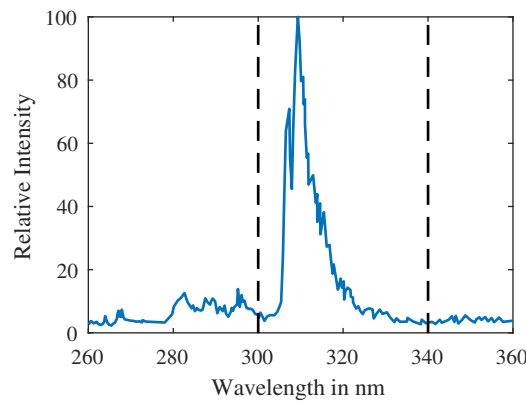


Figure 4.4: Emission spectrum of a H_2 -air co-flow flame, taken from Zhao et al. [205]. The dashed lines mark the cut-off frequency of the band-pass filter.

Table 4.3: Steam mass flow rates in the CTR A for the OH* recordings.

#	P_{th} in kW	$\dot{m}_{H_2O}$ in kg/h	Ω	T_{ad} in K	Re_{H_2O}
1	51	33.1	2.5	1966	10239
2	51	36.2	2.7	1890	11202
3	51	57.4	4.3	1505	17719
4	51	86.5	6.4	1215	26601
5	51	84.4	6.3	1230	26038
6	51	104.7	7.8	1098	32269

Table 4.4: Reactants mass flow rates in the CTR A for the recording of the operational maps.

#	P_{th} in kW	$\dot{m}_{H_2}$ in kg/h	$\dot{m}_{O_2}$ in kg/h	Re_{H_2}	Re_{O_2}
1	20	0.61	4.78	2425	1644
2	30	0.9	7.12	3604	2448
3	40	1.2	9.59	4789	3295
4	50	1.49	12.02	5979	4130
5	60	1.8	14.34	7199	4929
6	69	2.06	16.66	8267	5726

The operational maps were recorded in a separate measurement campaign with much more data points but without flame imaging. The flame shape was determined by manual observation. The steam mass flow rate was increased from a minimum of 20 kg/h to the cold blow-off of the flame, and the thermal power was varied in the six fixed levels, given in Tab. 4.4. The flame remained stoichiometric during the tests. The mass flow rates and temperatures were recorded for 120 s and time-averaged subsequently. In addition to Configuration III and VII, an operational map was recorded for Configuration I as well. All three are discussed in Sec. 4.4.

4.2.2 Heat Transfer Measurements

In general, there are three ways of heat transfer, namely thermal conduction, thermal convection, and radiation. The major part of the heat released in the combustion chamber is convected in the axial direction by the exhaust flow. However, the heat transfer in the radial direction to the chamber walls is of high interest to estimate the heat loss and the material load. Hence, the analysis in this thesis focused on the total heat transfer from the combustion gases to the combustion chamber wall, caused by both conduction and radiation. As an introduction to the topic, this section first gives a short literature review of heat transfer measurements at combustion chambers. The analysis of the heat transfer in the current thesis had two distinct goals:

- the measurement of the total heat transfer to evaluate the effectiveness of the combustion chamber cooling,
- and the determination of spatially resolved wall temperatures and Nusselt numbers inside the combustion chamber, which are relevant for the further combustor development and power scaling.

The methods for both parts are presented separately after the literature review.

Literature Review on Heat Transfer in Combustion Chambers

Different techniques to measure the heat transfer to the wall were reported in the literature. Patil et al. [137] studied the heat transfer of a cold combustor flow by heating the wall and recording its temperature with an IR camera. The results showed that the Nusselt number was very high at the end of the secondary recirculation zones and then declined quickly downstream. A similar result was found by Hedlund and Ligrani [81] for the flow in a swirl chamber.

The use of heat flux sensors is another option to determine heat transfer. For instance, Zhen et al. [206] measured the heat flux on a flame impingement plate at different nozzle-to-plate distances. Wu et al. [201] utilized water-cooled heat flux meters at the combustion chamber wall, and coaxial thermocouple heat flux gauges were employed by Jones et al. [93].

A further method was used by Back et al. [14, 15] to study the heat transfer of hot combustion exhaust to a water-cooled nozzle wall. They calculated the heat transfer coefficients from the coolant mass flow rate, the temperature increase, and the surface area. This steady-state calorimetric method was described in detail by Witte and Harper [196] and in the review of Bartz [18]. The same technique was utilized by Bělohradský et al. [22] to measure the heat transfer in a natural gas furnace combustion chamber.

As the application of this calorimetric technique [15, 18, 22] approach is straightforward, it was employed similarly for the measurements in the thesis. However, the current study used air instead of water the cooling flow.

Instead of natural gas and air, the combustor in the current work consumed H_2 and O_2 , which has a significant influence on the heat transfer. Bělohradský et al. [22] studied the impact of O_2 enrichment. They found a significant increase in the heat flux – up to 46 % – and contributed this to enhanced radiation due to the higher H_2O and CO_2 concentrations. Bäckström et al. [16] and Andersson et al. [8] came to the same conclusions by measuring the radiative heat transfer of oxygen-enriched flames optically. Baukal and Gebhart [19] additionally found that the radiant heat flux reaches its maximum value for stoichiometric flames with pure oxygen.

Zhen et al. [206] studied the effect of hydrogen enrichment up to 50 %_v to a premixed LPG flame. While hydrogen had a massive impact on the emissions and shifted the point of maximum local heat flux closer to the nozzle, it had only a minor effect on the total heat transfer rate, which was increased by 3 %.

The effect of simultaneous O_2 and H_2 enrichment to a laminar methane diffusion flame was studied by Wu et al. [201]. They found that the radiative heat flux increased with the oxygen level, but decreased with increasing H_2 addition due to lower CO_2 and soot concentrations. The total heat flux, on the other hand, increased for both enrichments.

These effects also apply to the current study, in which the oxidizer is pure O_2 and the flue gas almost pure H_2O . Cremers et al. [45] analytically and numerically discussed the heat transfer of such H_2/O_2 flames. Measurements of the heat transfer in rocket combustors, which are the primary application of these flames today, were reported by Droppers et al. [51] and Jones et al. [93]. Both studies aimed to provide databases for CFD validation.

In order to evaluate the heat transfer, the thermal properties of the materials must be known. The VDI-Wärmeatlas [192] provides a broad range of tabulated data, which has been extracted and interpolated for the actual temperature in the current work. The necessary values are given for some exemplary temperatures in Tab. 4.5. In the case of the reactants, H_2 and O_2 , the supply temperature is assumed to be 293 K. Accordingly, the specific enthalpies are $h_{\text{H}_2} = 3860 \text{ kJ/kg}$ and $h_{\text{O}_2} = 266.4 \text{ kJ/kg}$.

Table 4.5: Thermal properties for the experiments. These values are only examples to give an impression of the order of magnitude. In the analysis the values are interpolated from [116, 192] for the actual temperature.

Material						
Stainless steel 1.4571						
	Temperature		293	400	600	1000 K
	Thermal conductivity	λ_w	13	15.5	18.5	24.3 W/Km
Fused Silica						
	Temperature		293	400	600	1000 K
	Thermal conductivity	λ_{fs}	1.38	1.4	1.43	1.58 W/Km
Air						
	Temperature		293	350	400	450 K
	Thermal conductivity	λ_{air}	25.9	23.2	33.5	36.8 mW/Km
	Specific heat capacity	cp_{air}	1.01	1.01	1.02	1.02 kJ/KgK
Steam						
	Temperature		400	600	800	1000 K
	Thermal conductivity	λ_{H_2O}	26.8	46.4	70.3	97.3 mW/Km
	Specific heat capacity	cp_{H_2O}	2.01	2.03	2.15	2.29 kJ/KgK
	Specific enthalpy	h_{H_2O}	2730	3129	3546	3991 kJ/Kg

Total Heat Balance

In general, the total heat balance of a system compares the incoming and outgoing enthalpy fluxes. Thus the combustion chamber and the cooling duct were treated as black boxes, as shown in Fig. 4.5.

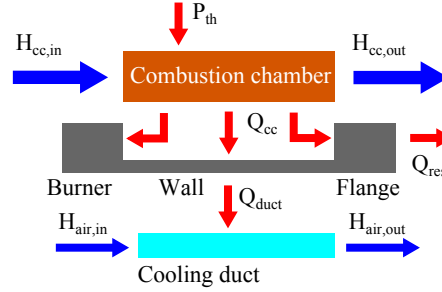


Figure 4.5: Schematic of the total heat transfer model.

In the case of the combustion chamber, the incoming energy was provided by the inlet flow and the heat release of the flame, while the outgoing includes the exhaust stream and the heat loss to the metal components. Hence the heat balance in the combustion chamber reads

$$P_{th} = \Delta \dot{H}_{cc} - \dot{Q}_{cc}, \quad (4.1)$$

with the enthalpy flux difference between the in- and outflow being

$$\Delta \dot{H}_{cc} = \dot{H}_{cc,out} - \dot{H}_{cc,in}. \quad (4.2)$$

At the inlet of the combustion chamber, the three incoming fluids with their specific

enthalpies $h_{\text{H}_2\text{O}}(T_{\text{plenum}})$, h_{H_2} , and h_{O_2} were combined to the inlet enthalpy flux $\dot{H}_{\text{cc,in}}$.

$$\dot{H}_{\text{cc,in}} = \dot{m}_{\text{H}_2\text{O}} h_{\text{H}_2\text{O}}(T_{\text{plenum}}) + \dot{m}_{\text{H}_2} h_{\text{H}_2} + \dot{m}_{\text{O}_2} h_{\text{O}_2} \quad (4.3)$$

The steam enthalpy was evaluated at the burner plenum temperature and the reactant enthalpies at room temperature.

For the calculation of the outlet stream, perfect combustion was assumed. Hence, the exhaust gas was treated as pure steam, and the enthalpy of the exhaust flow $\dot{m}_{\text{ex}} = \dot{m}_{\text{H}_2\text{O}} + \dot{m}_{\text{H}_2} + \dot{m}_{\text{O}_2}$ was determined at the exhaust temperature T_{ex} .

$$\dot{H}_{\text{cc,out}} = \dot{m}_{\text{ex}} h_{\text{H}_2\text{O}}(T_{\text{ex}}) \quad (4.4)$$

As $\Delta\dot{H}_{\text{cc}}$ and P_{th} were known from the BCs and the steam temperatures, Eq. 4.1 yielded the conductive heat loss of the combustion chamber $\dot{Q}_{\text{cc}}$.

The second black box was the cooling duct, which was purged by a co-flow of air. Here, the inputs were the incoming air and the heat conducted through the combustion chamber wall. The cooling duct was thermally insulated against the ambient surrounding. Thus the only output was the outflow of the cooling air. As a result, the heat flux through the combustion chamber wall was given by the enthalpy difference between the air in- and outlet:

$$\dot{Q}_{\text{duct}} = \Delta\dot{H}_{\text{duct}}, \quad (4.5)$$

with

$$\Delta\dot{H}_{\text{duct}} = \dot{m}_{\text{air}} (cp_{\text{air,out}} T_{\text{air,out}} - cp_{\text{air,in}} T_{\text{air,in}}). \quad (4.6)$$

The conductive heat losses of the plenum, the duct, and the flanges should be low due to the thermal insulation. Nevertheless, they were not negligible in the experiments because they disturbed the system's heat balance, i.e., $\dot{Q}_{\text{cc}} \neq -\dot{Q}_{\text{duct}}$. Hence, there was a residual heat flux

$$\dot{Q}_{\text{res}} = \dot{Q}_{\text{cc}} + \dot{Q}_{\text{duct}}. \quad (4.7)$$

The exhaust temperature used in Eq. (4.4) was measured by an unshielded thermocouple. For very high exhaust temperatures, it started to glow. Hence, the measured temperatures were corrected for the thermocouple's radiative heat loss using a model proposed by Příkopský [142]. The correction is shown in Fig. 4.6 for an exemplary flame at different steam dilution ratios.

Heat Transfer through the Wall

The spatially resolved wall temperatures and Nusselt numbers were estimated by a network model using the cell method. Therefore, the cooling duct, the wall, and the combustion chamber were divided into seven virtual cells with $L_{\text{cell}} = 20$ mm. Fig. 4.7 shows the network with its main quantities, which were determined from bottom to top in the analysis.

The temperature in the cooling duct T_{air} was measured at the boundaries of each cell j , as shown by the black lines at the bottom of Fig. 4.7. Hence, the temperature difference of each cell could be calculated. It further yielded the change of enthalpy Δh_j in the cells and

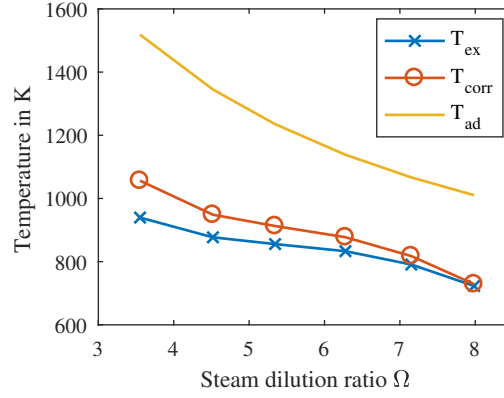


Figure 4.6: Correction of the thermocouple measurements in the exhaust.

the heat flux $\dot{Q}_j$ to the cooling duct in each cell.

$$\Delta h_j = c_{p,\text{air},j+1} T_{\text{air},j+1} - c_{p,\text{air},j} T_{\text{air},j} \quad (4.8)$$

$$\dot{Q}_j = \Delta h_j \dot{m}_{\text{air}} \quad (4.9)$$

The heat flux into the cooling duct $\dot{Q}_j$ equaled the amount of heat per cell conducted from the combustion chamber through the wall. The next step was to determine the wall temperature of the cells via the heat transfer coefficient α_j . Therefore, the modeling of the air wall interaction followed Chapter Gb of the VDI-Wärmeatlas [192].

$$\alpha_{\text{air},j} = \frac{1}{\pi d_{\text{duct},i} L_{\text{cell}}} \frac{\dot{Q}_j}{\Delta_{\ln} T_{\text{air},j}} \quad (4.10)$$

$$\Delta_{\ln} T_{\text{air},j} = \frac{(T_{\text{ow},j} - T_{\text{air},j}) - (T_{\text{ow},j} - T_{\text{air},j+1})}{\ln(T_{\text{ow},j} - T_{\text{air},j}) - \ln(T_{\text{ow},j} - T_{\text{air},j+1})} \quad (4.11)$$

The definition of α_j contains the logarithmic temperature difference $\Delta_{\ln} T_{\text{air},i}$, which depends on the outer temperature of the combustion chamber. Hence, if α_j is known, the

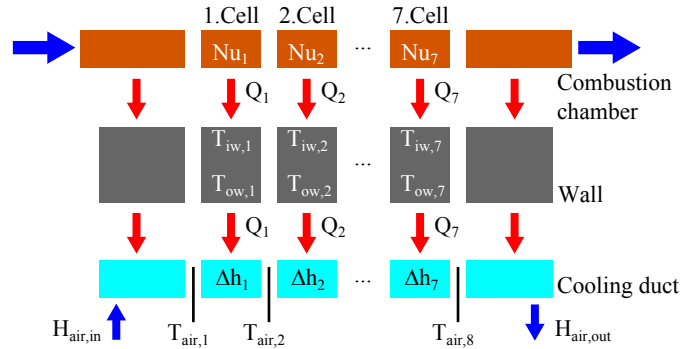


Figure 4.7: Schematic of the heat transfer network model.

wall temperature can be calculated by Eq. (4.10) and (4.11). To determine α_j , a Nusselt number correlation for the annular duct flow was used:

$$Nu_{\text{air},j} = \frac{\alpha_{\text{air},j}}{\lambda_{\text{air},j}} d_{\text{duct},h}, \quad (4.12)$$

with $\lambda_{\text{air},j}$ being the thermal conductivity and $d_{\text{duct},h}$ the hydraulic diameter of the duct (see Tab. 4.2). The Nusselt number does not only depend on the thermal properties but also on the flow.

There are Nusselt number correlations for turbulent and laminar flows in an annulus [139, 175, 192]. The Reynolds number of the flow in the duct was in the order of $Re_{\text{air}} = 1500$ –2000, which is below the critical one for duct flows. However, the Reynolds number of the inflow was much higher (≈ 20000). Moreover, at the beginning of the duct, the cooling air flowed through a slotted metal sheet, which generated additional turbulence. The turbulence increases the heat absorption significantly. A laminar model, therefore, led to unreasonable high wall temperatures and was not applicable. Instead, a turbulent flow was assumed in the cooling duct and the slots were taken as the reference to calculate the modified Reynolds number. Therefore, the bulk flow velocity at the slots was the characteristic velocity and $d_{\text{duct},s} = 35$ mm the characteristic length. This assumption yielded Reynolds numbers in the range of $Re_m = 3500$ –5000.

The formulation of the Nusselt number blends from the laminar to the turbulent correlation in the case of Reynolds numbers between 2300 and 10000. The Nusselt number for the annular duct then reads

$$Nu_{b,j} = (1 - \gamma_j) Nu_{l,j} + \gamma_j Nu_{t,j} \quad (4.13)$$

with the blending function $\gamma_j = (Re_{m,j} - Re_l)/(Re_t - Re_l)$. Both Nusselt numbers, the turbulent and the laminar, were adapted from the round tube by applying correction factors, which depend on the diameter ratio $a = d_{\text{duct},i}/d_{\text{duct},o}$. Gnielinski [70] reported the turbulent one with $Re_t = 10000$ as follows.

$$Nu_{\text{tube},j} = \frac{\frac{\xi_j}{8} Re_t Pr_{\text{air},j}}{k_1 + 12.7 \sqrt{\frac{\xi_j}{8}} \left(Pr_{\text{air},j}^{\frac{2}{3}} - 1 \right)} \left(1 + \left(\frac{d_{\text{duct},h}}{L_{\text{cell}}} \right)^{\frac{2}{3}} \right) \quad (4.14)$$

$$k_1 = 1.07 + \frac{900}{Re_t} - \frac{0.63}{1 + 10 Pr_{\text{air},j}} \quad (4.15)$$

$$\xi_j = (1.8 \log(Re_t k_2) - 1.5)^{-2} \quad (4.16)$$

ξ_j is the friction coefficient, which differs between the round tube and the annulus. Gnielinski [69] proposed the usage of the correction factor

$$k_2 = \frac{(1 + a^2) \ln a + (1 - a^2)}{(1 - a^2) \ln a}. \quad (4.17)$$

Finally, the turbulent Nusselt number for the annulus reads

$$Nu_{t,j} = 0.75 a^{-0.17} Nu_{\text{tube},j}. \quad (4.18)$$

According to [192], the laminar Nusselt number was evaluated at $Re_1 = 2300$ in three steps, which accounted for different axial phases of the flow in the duct. Since the cooling section was relatively short, all three had to be considered.

$$Nu_{l1,j} = 3.66 + 1.2 a^{-0.8} \quad (4.19)$$

$$Nu_{l2,j} = 1.615 \left(1 + 0.14 a^{-0.5}\right) \left(Re_1 Pr_{air,j} \frac{d_{duct,h}}{L_{cell}}\right)^{\frac{1}{3}} \quad (4.20)$$

$$Nu_{l3,j} = \left(\frac{2}{1 + 22 Pr_{air,j}}\right)^{\frac{1}{6}} \left(Re_1 Pr_{air,j} \frac{d_{duct,h}}{L_{cell}}\right)^{\frac{1}{2}} \quad (4.21)$$

The mean laminar Nusselt number summed up to

$$Nu_{l,j} = \left(Nu_{l1,j}^3 + Nu_{l2,j}^3 + Nu_{l3,j}^3\right)^{\frac{1}{3}}. \quad (4.22)$$

The medium properties in the above equations were determined at the mean air temperature $\bar{T}_{air,j} = (T_j + T_{j+1})/2$. A correction of the temperature influence for the Nusselt number of the intermediate range $Nu_{b,j}$ (Eq. (4.13)) was applied

$$Nu_{air,j} = \left(\frac{\bar{T}_{air,j}}{T_{ow,j}}\right)^n Nu_{b,j} \quad (4.23)$$

with $n = 0.45$ [192]. This correction required the unknown wall temperature $T_{ow,j}$. Thus, the Eq. (4.10)-(4.12) and (4.23) were solved iteratively and yielded the outside temperature of the combustion chamber wall. With the final Nusselt number of the flow, all necessary quantities from the cooling duct were known: $\dot{Q}_j$, $T_{ow,j}$, and $Nu_{air,j}$.

The second step evaluated the heat conduction through the wall. The temperature difference $\Delta T_{w,j} = T_{iw,j} - T_{ow,j}$ was calculated from the heat flux as

$$\Delta T_{w,j} = \frac{\dot{Q}_j}{2\pi L_{cell} \lambda_{w,j}} \ln\left(\frac{d_{duct,i}}{D}\right) \quad (4.24)$$

with D being the inner and $d_{duct,i}$ the outer diameter of the combustion chamber wall. The thermal conductivity $\lambda_{w,j}$ of the steel (Tab. 4.5) was taken at the outer wall temperature.

At last, Eq. (4.25) determined the Nusselt number inside the combustion chamber from the steam's heat transfer coefficient $\alpha_{H_2O,j}$ (Eq. (4.26)).

$$Nu_j = \frac{\alpha_{H_2O,j} D}{\lambda_{H_2O,j}} \quad (4.25)$$

$$\alpha_{H_2O,j} = \frac{1}{\pi D L_{cell}} \frac{\dot{Q}_j}{\Delta_{\ln} T_{H_2O,j}} \quad (4.26)$$

In these equations, the temperature distribution in the combustion chamber is required to determine the thermal conductivity $\lambda_{H_2O,j}$ and the logarithmic average temperature

$\Delta_{\ln} T_{\text{H}_2\text{O},j}$. However, the temperature distribution in the combustion chamber was not measured because the adiabatic flame temperatures of up to 2500 K were too high for any thermocouple. Moreover, the actual temperature distribution in the first part of the combustion chamber was far from homogeneous. As it generally depends on the wall distance [26], it does not serve as a useful reference. Instead, a reference temperature can be set manually. Even though, choosing different reference temperatures yields different Nusselt numbers, the Nusselt numbers of different experiments can still be compared, as long as the same reference is taken. In the best case, the reference temperature should be easy to determine. To illustrate this issue, two approaches to set this reference are compared in the results section.

The first approach to set the reference was to assume the adiabatic flame temperature in the entire combustion chamber, which is similar to Lefebvre and Herbert [115], who used a flame bulk temperature. In this case, the logarithmic temperature difference $\Delta_{\ln} T_{\text{H}_2\text{O},j}$ was not defined and replaced by $T_{\text{iw},j} - T_{\text{ad}}$ according to L'Hospital's rule for limits of indeterminate forms.

Secondly, a linear temperature distribution was assumed using the boundary conditions, which were the mixing temperature T_{mix} at the inlet and the exhaust temperature T_{ex} 60 mm downstream of the combustion chamber. In between, the adiabatic flame temperature was set to the flame position x_f (see Fig. 4.17). The gas temperature inside the combustion chamber was interpolated linearly between these values. The model is shown in Fig. 4.8, together with $T_{\text{H}_2\text{O},j}$ at the boundaries of the evaluation cells.

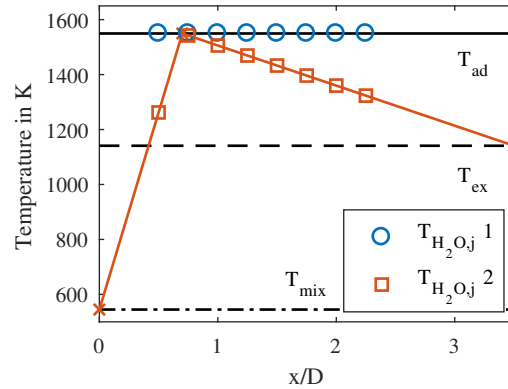


Figure 4.8: Two assumptions for the temperature in the combustion chamber: constant adiabatic flame temperature (circles) and a linear model (squares).

4.3 Flame Types and Shapes

The flow field measurements, presented in Chapter 3, showed two distinct types of flow fields: a low-swirl axial jet and a high-swirl expanding jet featuring VB. The huge difference in the flow characteristics of both types is expected to have a significant influence on the flame structure and position. Indeed, two flame types can be identified from the OH* recordings, exemplarily shown in Fig. 4.9.

The left contour illustrates the OH* intensity for a flame featuring a low steam dilution ratio. This flame type is considered as a jet flame throughout the thesis. On the right side, Fig. 4.9 shows a swirl-stabilized flame with a high level of steam dilution. Please note that the intensities in Fig. 4.9 were normalized by the maximum intensity of each image. The

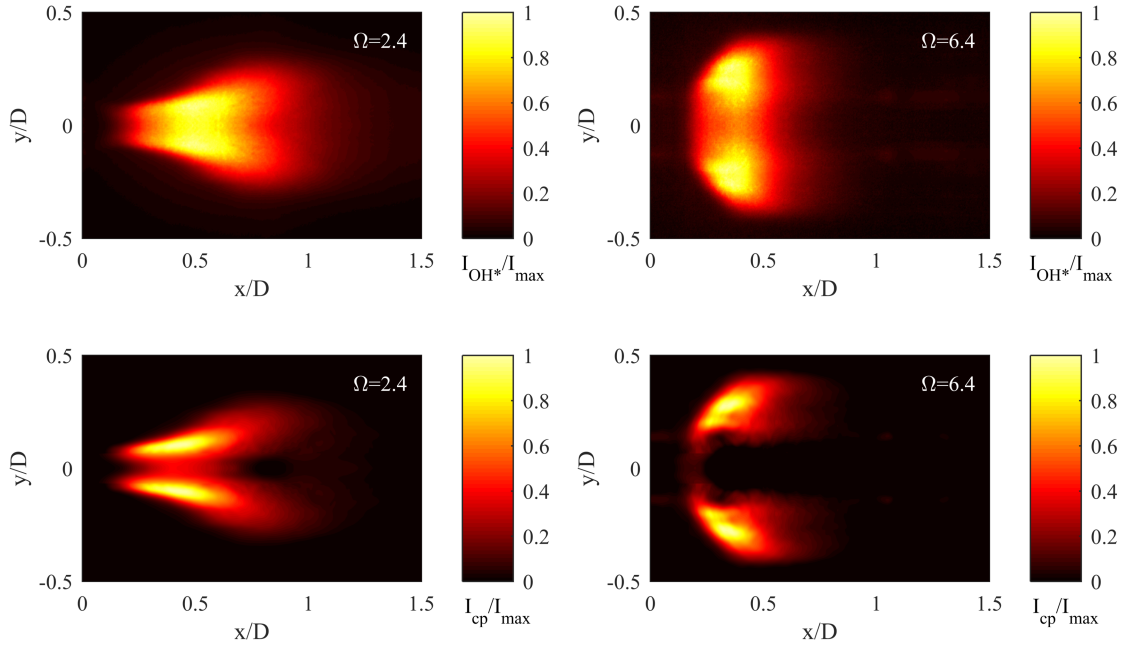


Figure 4.9: Top row: Relative line-of-sight OH* intensity of Configuration VII for two levels of steam dilution. Please note that the absolute intensity is about one order of magnitude higher in the left case. Bottom row: Relative OH* intensity of Configuration VII in the center plane of the combustion chamber. It is calculated by an inverse Abel transformation.

absolute intensity is about one order of magnitude higher for the jet flame.

The top row of Fig. 4.9 gives the line-of-sight integration of the OH* intensity through the round combustion chamber. However, to compare the flame shape and position with the PIV and PLIF data, the intensity distribution on the center plane would be much more suited. Hence, an inverse Abel transformation [168] is applied, which takes advantage of the rotational symmetry of the flame and yields a deconvolution of the intensity image. The OH* chemiluminescence intensity in the center plane of the combustion chamber is shown in the bottom row of Fig. 4.9 for both flame types. They are discussed separately in the following subsections. The transition between these modes is discussed afterwards in Sec. 4.4.

4.3.1 Jet Flame

The line-of-sight integrated OH* chemiluminescence image of the jet flame in the top left of Fig. 4.9 shows a high intensity around the burner's central axis, which spreads weakly in the radial direction. However, the deconvoluted intensity has its maximum not on the central axis but at $y/D=0.1$. Actually, the intensity at $y/D=0$ is very low, which indicates a less reacting zone at the centerline of the combustion chamber. This effect is caused by the H₂ jet on the central axis.

The OH* intensity distribution in the symmetry plane can be directly compared with the flow and mixing field. In Fig. 4.10, the relative, deconvoluted intensity field is superposed by vectors of the axial and radial velocity from the WT (see Sec. 3.3.1). The solid white line indicates zero axial velocity.

The vectors show that the flow expands in the radial direction from approximately $0.4D$ onward. As expected for the jet flame, the reaction zone follows exactly the streamlines.

The isothermal WT measurements seem to be an excellent indicator for the reacting flow field in this case. In addition to the contour, profiles of the OH* intensity and the velocity are given in Fig. 4.12 (left) together with the turbulence intensity. The profiles point out that the flame is located close to the peak velocity, which indicates that the flame is not stabilized aerodynamically because such flames would be located more in the SLs of an IRZ or downstream of a bluff body [39].

This statement is underlined by the comparison of the OH* chemiluminescence with the reactant's mixture fractions (see Sec. 3.4), depicted in Fig. 4.11. The green contour lines represent H₂ concentrations of 10 %, 30 %, 40 %, and 50 % of the injected H₂. Here, higher concentrations lie on the inner side of the contour lines. The same holds for the O₂ distribution, symbolized by the blue contour lines (no 10 %). The figure shows that the flame

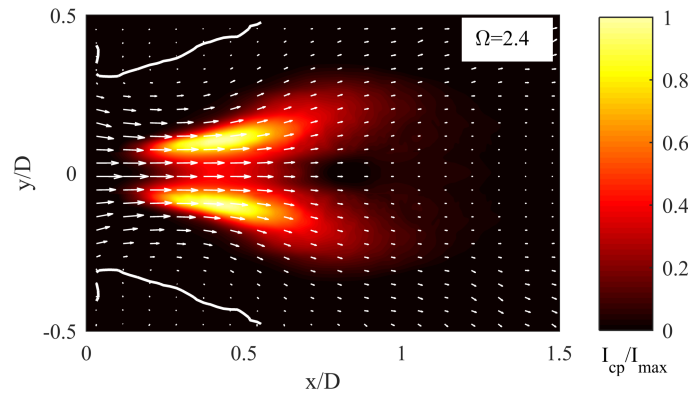


Figure 4.10: Relative OH* intensity of Configuration VII in the center plane of the combustion chamber, superposed by vectors and the stagnation line of the velocity field (WT, $\Omega = 2.6$).

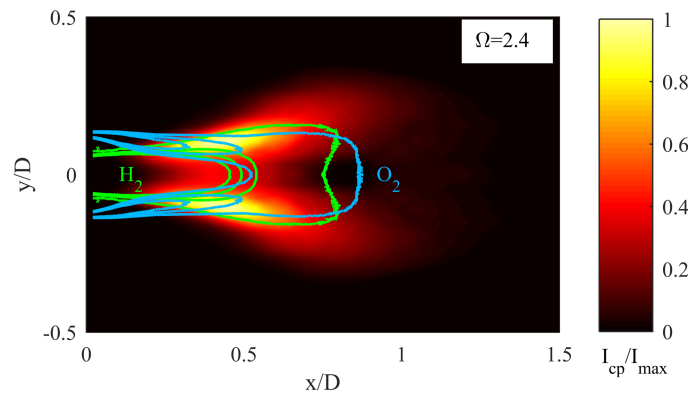


Figure 4.11: Relative OH* intensity of Configuration VII in the center plane of the combustion chamber, superposed by contour lines of H₂ (white) and O₂ (green) concentration (WT, $\Omega = 1.9$). The contour level represents 10 % (H₂ only), 30 %, 40 %, and 50 % of the injector concentration.

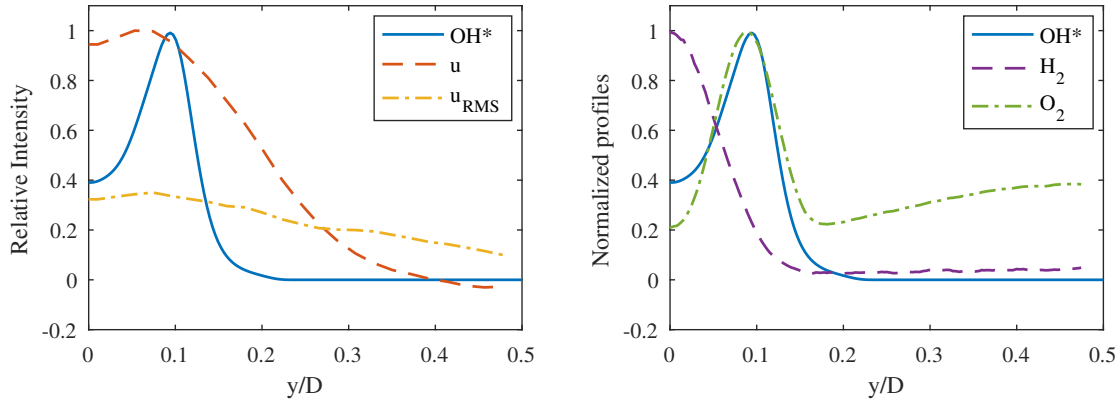


Figure 4.12: Relative OH* intensity profile of a jet flame (Configuration VII) at $x/D = 0.35$ are compared to the WT results. Left: Profiles of the normalized axial velocity and turbulence intensity. Right: Profiles of relative H₂ and O₂ concentration.

starts burning where the 40 % contour lines of H₂ and O₂ touch each other, i.e., at the point of a stoichiometric mixture ratio. Moreover, the figure reveals that the main reaction zone lies within the 10 % contour line of the H₂. Outside of this envelope, either the concentration is too low for combustion or the H₂ is already consumed by the upstream flame.

The profiles at $x/D = 0.35$ in Fig. 4.12 show that the inner gradient of the OH* intensity, i.e. the flame front, aligns with the intersection of the H₂ and O₂ concentration, where the mixture is stoichiometric. This behavior indicates that the flame is stabilized by the mixture field like a non-premixed flame, in which the reaction zone is fixed close to the region of a stoichiometric reactant ratio [194]. The fact that the flame is slightly shifted to the oxygen side might be explained by the underestimation of the molecular diffusivity of the hydrogen, as discussed in Sec. 3.2.2.

Even though the flame has diffusion flame characteristics, the reactant concentration differs from the typical one in Fig. 2.12 because there is no homogeneous distribution on the fuel nor the oxidizer side. Thus, it can be assumed that the flame front is locally non-stoichiometric, which might reduce combustion efficiency.

4.3.2 Swirl-Stabilized Flame

Increasing the steam dilution ratio leads to a transition in the flow field, as discussed in Chapter 3. The intensified swirl enables VB and forms a large IRZ in the center of the combustion chamber, which strongly affects the flame position. The integrated OH* chemiluminescence in the top row of Fig. 4.9 reveals that the flame is more compact in the axial direction but spreads radially. Thereby, the flame shape follows the flow vectors in Fig. 4.13, which point around the IRZ. The maximum heat release is located next to the stagnation line in the SLs. Inside of the IRZ, symbolized by the solid white line in Fig. 4.13, the heat release is very weak. Only close to the stagnation line, a weak glow is visible due to temporal fluctuations of the IRZ.

Fig. 4.14 shows the swirl flame superposed with H₂ and O₂ contours. Compared to the jet flame, the contour levels are lower (10 % (H₂ only), 15 % 20 %, and 30 %) because the increased steam dilution reduced the reactant concentrations. The contour lines intersect already upstream of the region of maximum heat release. This indicates that the reactants are partially premixed before combustion, which is a characteristic of swirl-stabilized flames [89].

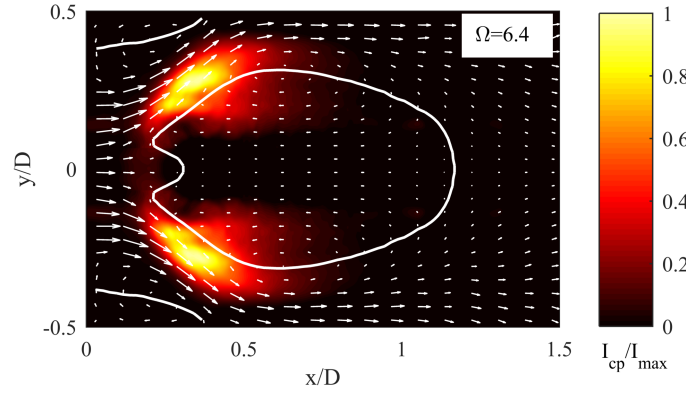


Figure 4.13: Relative OH* intensity of Configuration VII in the center plane of the combustion chamber, superposed by vectors and the stagnation line of the velocity field (WT, $\Omega = 6.4$).

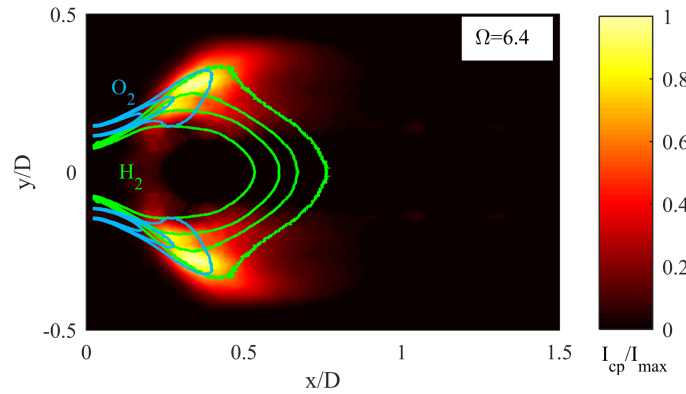


Figure 4.14: Relative OH* intensity of Configuration VII in the center plane of the combustion chamber, superposed by contour lines of H_2 (white) and O_2 (green) concentration (WT, $\Omega = 7.4$). The contour levels represent 10 % (H_2 only), 15 % 20 %, and 30 % of the injected O_2 .

Please note that the steam dilution ratio of both data sets does not match exactly due to a lag of data (OH*: $\Omega = 6.4$, PLIF: $\Omega = 7.4$). However, the flame topology does not change since both flames are in the swirl mode, and Ω is significantly above the transitional range.

The comparison of Fig. 4.13 and 4.14 shows that the H_2 contour line overlaps the stagnation line of the IRZ. H_2 is brought into the IRZ by turbulent transport. Nevertheless, no downstream convection of H_2 will appear along the centerline due to the steady-state backflow, as discussed in Sec. 3.3.1. Hence the H_2 is not transported through the IRZ but to the reaction zone in the SLs.

As for the jet flame before, Fig. 4.15 presents profiles of the OH* chemiluminescence, the axial velocity, the turbulence intensity (left), and the species concentrations (right). The maximum heat release is located in the inner SL, which supports the hypothesis of

swirl-stabilization. The flame automatically adapts its position to match the turbulent flame speed and the flow velocity.

The distribution of the O_2 concentration aligns well with the heat release zone. As stated before, the reactants cannot be transported along the centerline and are pushed outwards instead. Nevertheless, the reactant concentrations are small between the reaction zone and the wall. The reactants are kept away from the wall by the dilution from the outer steam duct. Thus, all reactants are transported through the hot zone of combustion and are expected to be burned. Additionally, the steam acts as an insulation layer and prevents the flame from getting quenched by the heat loss at the cold combustor walls.

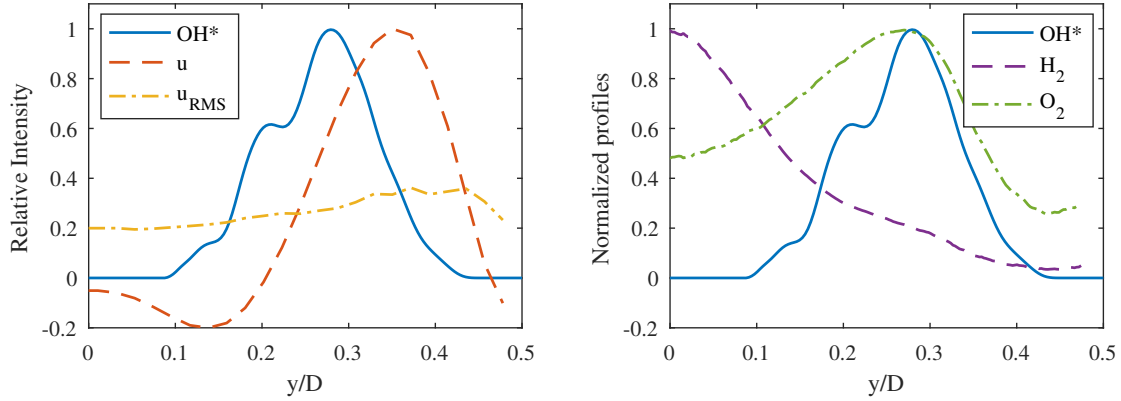


Figure 4.15: Relative OH^* intensity profile of a swirl-stabilized flame (Configuration VII) at $x/D = 0.35$ are compared to the WT results. Left: Profile of the normalized axial velocity and turbulence intensity. Right: Profiles of relative H_2 and O_2 concentration.

4.3.3 Influence of the Steam Dilution and the Swirl Number

In order to evaluate the influence of the steam dilution ratio on the flame shape, the following characteristics were analyzed: the total OH^* intensity, the axial flame position, and the flame length and angle.

The total OH^* chemiluminescence intensity was calculated by integrating the time-averaged intensity over the combustion chamber. It is shown in Fig. 4.16 (left) over the steam dilution ratio for Configuration III and VII. The integrated OH^* chemiluminescence is supposed to depend monotonically on the mean heat release rate of the flame, which is expected to decrease linearly with increasing flame dilution [111]. In fact, the measured OH^* intensity in Fig. 4.16 decreases linearly with Ω as well, and seems to be a good indicator for the heat release in this case. As the thermal power is kept constant, this decline is not related to changes in the fuel energy content. Instead, the higher steam mass flow linearly raises the transport rate of the reactants and reduces the adiabatic flame temperature (see Fig. 4.20).

The integral intensities of both burner configurations align well, showing that it is not affected by small changes of the swirl number in this burner. Lauer and Sattelmayer [112] reported that, among others, the turbulence intensity and the strain rate have a strong influence on the OH^* intensity. However, the flow field variations at constant Ω between Configuration III and VII are too low to influence the integrated intensity. In fact, the level of turbulence is similar in both cases, as shown in Fig. 4.16.

The axial flame position x_f is estimated as the distance between the combustion chamber

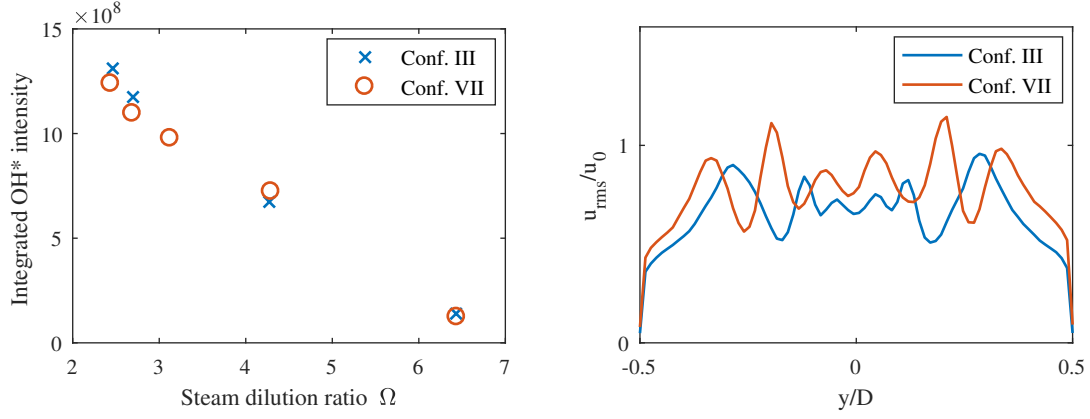


Figure 4.16: Left: total integrated OH* intensity of the flames in dependence of the steam dilution ratio for Configuration III and VII. Right: RANS turbulence intensity profiles at $x/D = 0.35$ and $\Omega = 4.2$.

inlet and the center of gravity of the OH* intensity distribution [184].

$$x_f = \frac{\int x I_{OH^*} dV}{\int I_{OH^*} dV} \quad (4.27)$$

On the left of Fig. 4.17, the position is plotted over the steam dilution ratio. Especially in the case of Configuration III, the data reveals that the flame position is clustered at two distinct positions (0.63 and 0.75 D). They are marked in as dashed lines in Fig. 4.17 (left) and stem from the two flame types. A similar behavior was previously reported by Terhaar [184] for different flame types generated by a variation of H₂ enrichment and steam dilution. For Configuration VII, the bifurcation is not as clear, but a switching point can be suspected here as well. However, the transition takes place at a lower steam dilution ratio, and the position is slightly more upstream for both flame types (0.54 and 0.72 D) due to the increased spreading of the flow, as discussed in Sec. 3.3.

Plotting the same data points over the estimated swirl number (Eq. (3.35)) indicates that an increasing swirl intensity shifts the flame upstream for the swirl-stabilized flame, while it

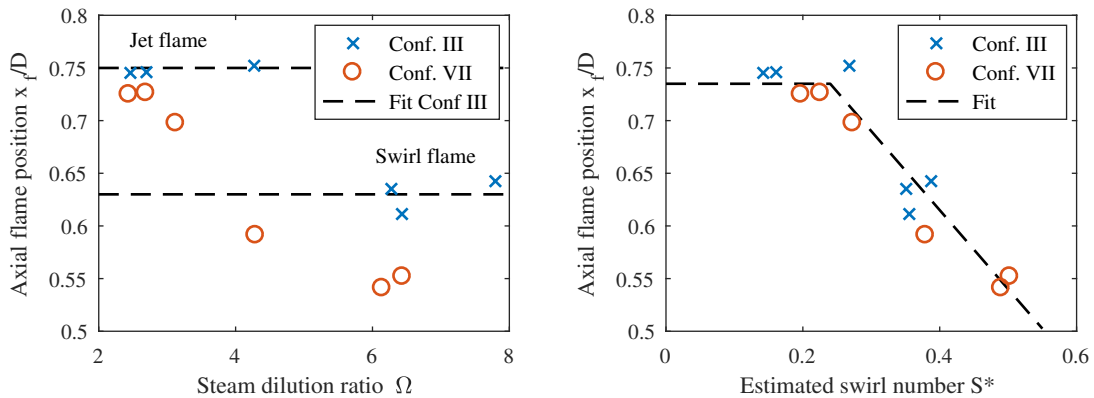


Figure 4.17: Axial flame position by the center of gravity of the OH* intensity in dependence of the steam dilution ratio (left) and the estimated swirl number (right) for Configuration III and VII.

does not affect the jet flame's position. This behavior aligns with the influence of the swirl intensity on the position of the IRZ (see Sec. 3.3.1). The dashed line in Fig. 4.17 (right) shows a fitted model, which is used in Sec. 4.5 to estimate the flame position depending on the swirl number.

Further characteristics are extracted from the deconvoluted OH^* intensity distribution, which is shown as single contour lines (0.1, 0.3, 0.5, 0.7, 0.9) in Fig. 4.18. From this distribution, the local OH^* intensity maximums in the radial direction are taken at each axial position. Linear regression is applied to those values lying inside of the 0.3 contour level. This threshold is a good trade-off since lower values include regions of very low OH^* intensity with uncertainties from the deconvolution, while higher values reduce the signal-to-noise ratio because fewer points of the flame are evaluated in the regression.

The regression coefficients give the axis of the flame, as depicted in Fig. 4.18. The flame axis and the centerline of the combustion chamber enclose the flame angle α_f . The flame length L_f is considered as the part of the flame axis with a relative intensity larger than 30 %. Thus it describes the length of the heat release zone. This definition is not universal, but it has proven useful in this thesis.

At low steam dilution ratios in the jet mode, the flame length is nearly constant, while it decreases significantly with enhanced steam dilution, as shown in Fig. 4.19 (left). This reduction is counter-intuitive because higher steam dilution causes an increased bulk flow velocity and a lower flame speed. Both effects should lengthen the flame. However, the flame length reduction can be attributed to the fact that the increased swirl intensity improves the turbulent mixing. This effect is emphasized in the swirl-stabilized mode due to the deflection of the flow around the IRZ. Here, the H_2 is mixed with the O_2 not only by turbulent diffusion but is convectively transported into O_2 -rich regions.

Apart from the declining trend, there is an exceptional point at $\Omega = 4.2$, where Configuration III features the longest of all measured flames. This flame is considered in the transition region between both flame types. Downstream of the area of maximum OH^* intensity, there is a plateau with approximately 60 % relative intensity, which enlarges the flame length artificially in this case.

At Configuration VII, the flame length declines with Ω as well. The flame is shorter than

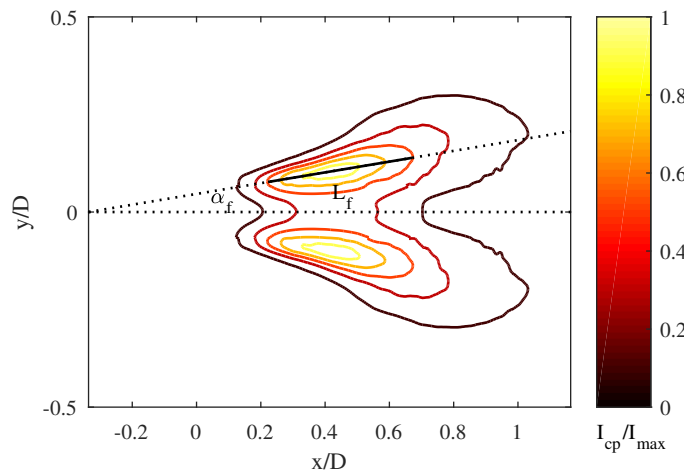


Figure 4.18: Contours of the deconvoluted OH^* intensity (Fig. 4.9 bottom left) at the levels 0.1, 0.3, 0.5, 0.7 and 0.9. Additionally, definitions of the flame length L_f and flame angle α_f are given.

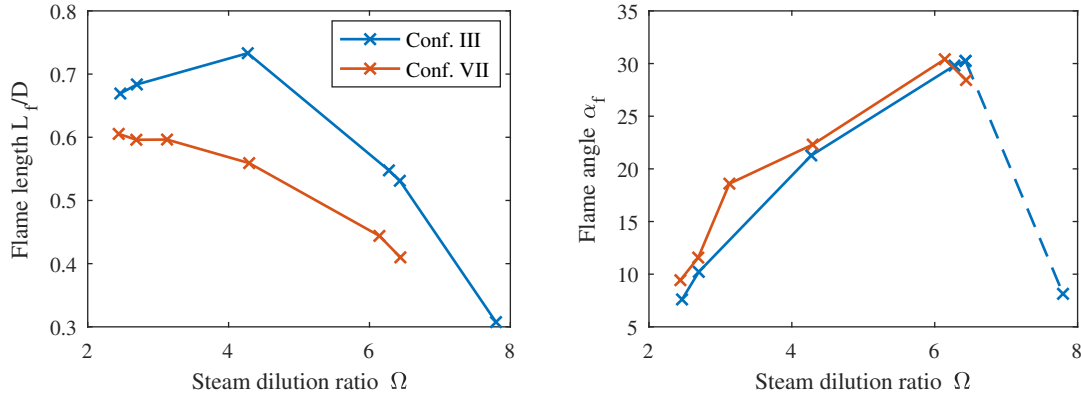


Figure 4.19: Flame length and angle in dependence of the steam dilution ratio for Configuration III and VII.

for Configuration III at all investigated steam dilution ratios, which goes along with the observed upstream shift due to the higher swirl number. In sum, the flame becomes more compact with enhanced swirl intensity and, hence, also with increased steam dilution.

As the flame shortens, its angle grows linearly with the steam dilution ratio, as shown in Fig. 4.19 (right). This effect can be explained by the enhanced swirl intensity that forces the flow to diverge at the combustion chamber inlet, as described in Sec. 3.3. As soon as the IRZ is established, the gas has to flow around it. Fig. 3.8 previously showed that the IRZ moves upstream and that its diameter rises at higher steam dilution ratios, which means that the diverging angle of the flow must grow as well. The flame simultaneously follows the streamlines of the flow field, as pointed out in Fig. 4.13. Accordingly, the flame angle increases from 5° to 30° for Configuration III. At the highest steam dilution ratio, the flame angle determination fails because the flame reveals a round instead of a longish shape.

Configuration VII shows a slightly lower flame angle because it features a higher swirl intensity. However, the effect is small and depends on the chosen threshold of the OH^* intensity. For lower values than 0.3, the difference between both curves in Fig. 4.19 increases. The effect vanishes or is even reversed for higher values, but in this case the regression is based on fewer points and less robust, especially for the short flames at high dilution ratios.

4.4 Operational Maps

The current section studies the range in which the combustor can be technically operated and under which conditions. Further operational restrictions will be discussed in Sec. 4.5 regarding the heat load on the combustion chamber walls, and in Chapter 5 regarding the combustion efficiency.

Two flame types were found in the analysis of the OH^* recordings: The jet and the swirl-stabilized flame. In order to evaluate the burner's operational range, three burner configurations (I, III, VII) were operated at different levels of steam dilution and thermal power. Please note that CTR A was used for this study. Therefore, the steam in the burners plenum is saturated, not superheated.

Even though the burner was tested for an extensive range of steam dilution ratios, the actual operation in real industrial processes is determined by the boundary conditions of the respective process. The main parameter here is the turbine inlet temperature. Figure 4.20 shows that the adiabatic flame temperature, calculated using *Cantera*, is a unique function

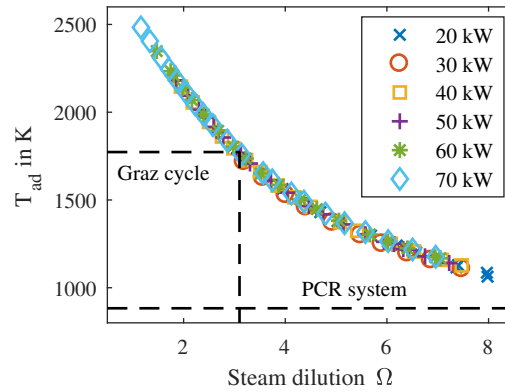


Figure 4.20: Dependency of the adiabatic flame temperature T_{ad} and the steam dilution ratio Ω for different levels of thermal power.

of the steam dilution ratio and does not depend on thermal power. This enhances the operational flexibility because the thermal power, i.e., the power output of the system, can be varied without changing T_{ad} as long as Ω is kept constant. As orientation, the dashed lines in Fig. 4.20 depict the combustor outlet temperatures of the PCR system and the Graz cycle, which were discussed as possible applications in Sec. 2.1 and 2.2.

In general, there are physical boundaries of the combustor's operating range. On the low dilution side, the boundary is set by the thermal load of the combustion chamber walls, as discussed in Sec. 4.5. On the high dilution side, the flame will distinguish by cold blow-off if the adiabatic flame temperature is too low. Additionally, Chapter 5 will show that combustion efficiency drops for $\Omega > 6.5$, which further limits the operational range. With these boundaries, the low combustor outlet temperature of the PCR system can not be reached by the adiabatic flame temperature. Nevertheless, the combustor can be used for this application because the exhaust can be further diluted by cold steam downstream of the combustor to reach the aimed outlet temperature.

For Configuration I, the operational map is shown at the top of Fig. 4.21 for thermal power levels of 20-70 kW. The flame is ignited at low steam dilution ratios in the jet flame regime depicted in the figure by filled triangles.

Increasing the steam dilution sets the flame in an unstable mode. In this situation, the flame completely detaches from the burner and propagates downstream through the combustion chamber. It anchors near the end of the combustion chamber and spreads over the whole cross-section. This mode is associated with high fluctuations of the flame position, as well as pressure pulsations, and should be avoided in any case in real operation because the flame is not in its intended position and could damage downstream components, e.g., the first turbine stage. Moreover, pressure pulsations can reduce the lifetime of the combustion chamber. If the steam dilution is further increased, the flame jumps upstream again and switches into the swirl-stabilized mode. The instability of the flame is also observed at the reverse transition from the swirl-stabilized to the jet flame. In Fig. 4.21, the unstable operation points are plotted as unfilled squares, while the swirl-stabilized flame type is plotted as filled dots. Please note that unstable transition occurred only in the experiments with the saturated steam. Superheating the steam upstream of the plenum enabled a stable transition even for Configuration I.

The steam dilution ratio, at which the transition takes place, depends on the thermal power level. This finding suggests that the transition is not solely driven by the swirl intensity

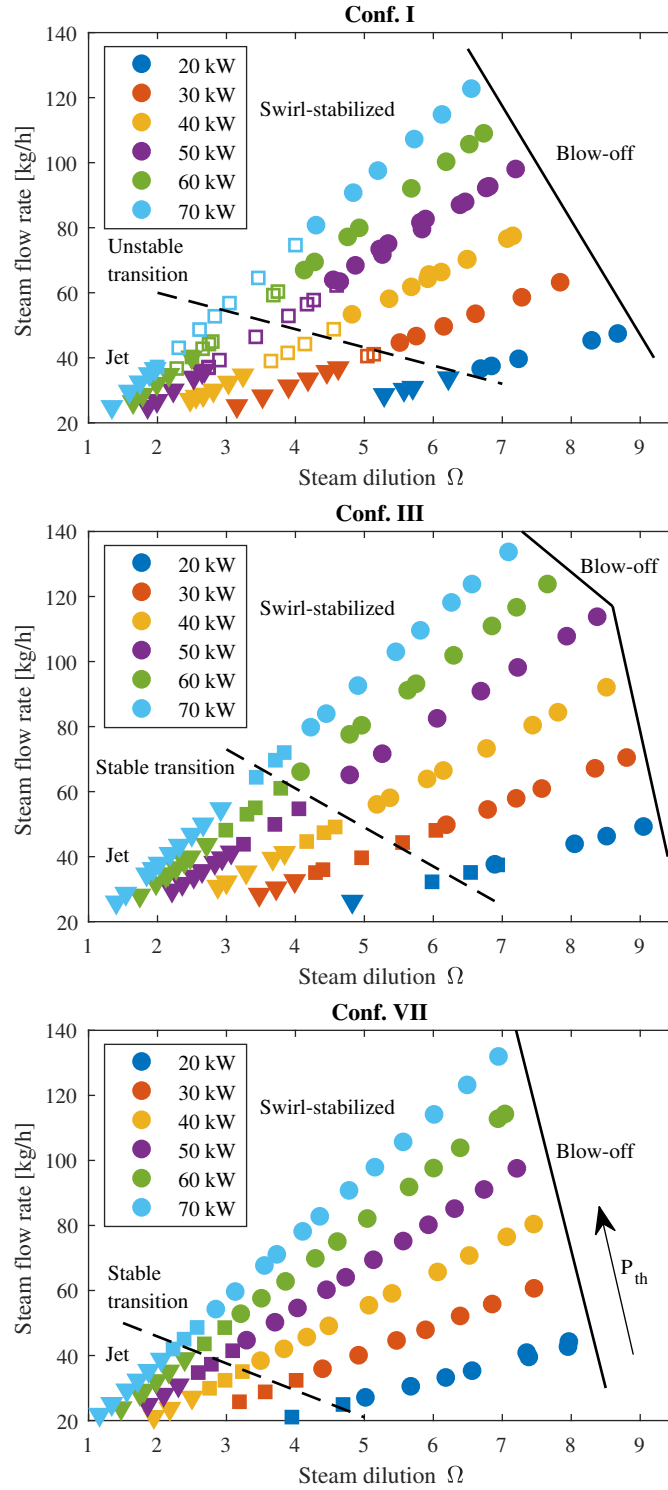


Figure 4.21: Operational maps of three burner configurations with the flames types: Jet flame (triangle), swirl-stabilized flame (dot), stable transition (filled square) and unstable transition (square).

of the flow, which is supposed to be constant for fixed Ω (see Fig. 3.16). Instead, it seems that the increase in heat release is shifting the flame transition towards lower swirl intensity. Konle and Sattelmeyer [98] investigated the influence of the heat release on VB in the context

of flashback prevention. They found that the combustion-induced production of azimuthal vorticity can strengthen the recirculation. Oberleithner et al. [134] showed the evolution of the vorticity field during the onset of VB. The negative azimuthal vorticity grows along the central axis as the swirl number increases. The enhanced vorticity production at higher thermal power might, therefore, reduce the critical swirl number for VB onset, explaining the observed effect.

In the swirl mode, the flame burns very stably over a broad range of steam dilution ratios. Diluting the flame even stronger lowers the adiabatic flame temperature until the limit of flammability is reached. In this case, cold blow-off appears, usually announced by very low and fluctuating light emission. The blow-off limit is marked with the solid black line in the figure. It depends on the power level as it decreases to lower Ω for higher thermal power. Cavaliere et al. [36] stated that blow-off occurs on a constant inverse Damköhler number

$$\frac{1}{Da} = \left(\frac{u_o}{D} \frac{\nu}{s_L^2} \right)^{0.5}. \quad (4.28)$$

Here, the bulk flow velocity u_0 rises linearly with the thermal power at constant Ω . The laminar flame speed s_L , as the counterpart, increases only slightly due to an enhanced preheating of the reaction zone [104]. Consequently, if the thermal power is increased, the steam dilution must be reduced to remain $1/Da$ constant, which explains the earlier blow-off.

Configuration III and VII provide an increased swirl intensity compared to Configuration I over the entire range of Ω . Indeed, Fig. 3.16 shows that the swirl number is not just elevated at higher configuration numbers, but also grows faster with the steam dilution ratio. The steeper gradient of the swirl number leads to a shorter transition phase from the jet to the swirl-stabilized flame. As a result, Configuration III and VII do not show any instability at all during this transition. Instead, they perform a utterly stable transition from the jet to the swirl-stabilized mode and vice versa without lift-off or hysteresis. The transitional operation points are marked as filled squares in Fig. 4.21 (middle and bottom). An example of such a transitional flame is given in Fig. 4.22 with the vectors of the velocity field from the WT. One can see that the flame already spreads with the flow, but the swirl intensity is still too low to enable VB.

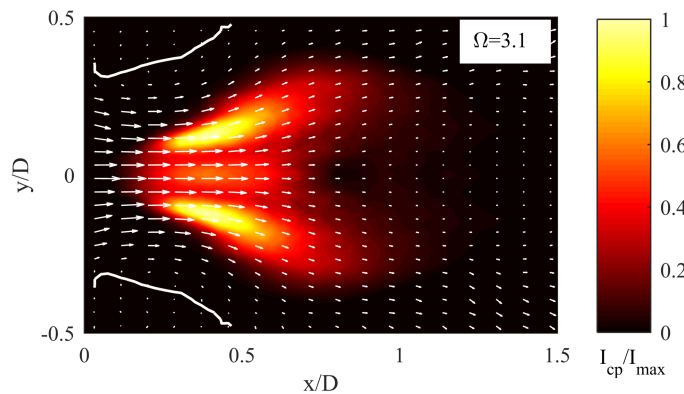


Figure 4.22: Relative OH* intensity of a transitional flame (Configuration VII, $\Omega = 3.1$) in the center plane of the combustion chamber, superposed by vectors and the stagnation line of the velocity field (WT, $\Omega = 3.0$).

Since the transition is stable at Configuration III and VII, it is hard to define a strict boundary as the flame fades from one mode to the other. The range of Ω , in which the transition occurs, is approximately the same for Configuration I and III. At Configuration VII, on the other hand, it moves towards much lower Ω . This down-shift enables a broader operation in the swirl-stabilized mode, which is beneficial because the flame shape variations with Ω are less strong if the flame remains in the same mode.

On the other hand, Configuration III reveals a larger blow-off limit, which means that the flame could be operated at slightly lower adiabatic flame temperatures. However, this effect is of minor importance because the flame will not be operated close to blow-off conditions due to safety reasons and the low combustion efficiency for $\Omega > 6.5$, as discussed in Chapter 5.

4.5 Heat Transfer

The results of the heat transfer measurements are divided into two parts: the total heat balance of the combustion chamber system and the evaluation of the heat transfer through its wall.

4.5.1 Total Heat Balance

The total heat balance of the combustion system is shown for Configuration III in Fig. 4.23. In this example, the flame's heat input is fixed to $P_{th} = 30$ kW. While flowing through the combustion chamber, the flow's enthalpy is increased about $\Delta\dot{H}_{cc}$ (Eq. (4.2)). Figure 4.23 (left) shows this enthalpy gain as the lower field (yellow). The difference between the enthalpy gain and the energy, provided by the flame, corresponds to the total heat loss $\dot{Q}_{cc}$ of the combustor system. It is shown as the red field in the left plot of Fig. 4.23 and as the total area in the right plot. If the system boundaries would be adiabatic, the heat flux to the cooling duct $\dot{Q}_{duct}$ should be equal to the heat loss of the combustion chamber, i.e., $\dot{Q}_{duct} \approx -\dot{Q}_{cc}$. However, Figure 4.23 (right) shows that the air in the cooling duct absorbs only the part of $\dot{Q}_{cc}$ that is shown by the bottom field (blue). In the actual system, further heat losses exist via heat conduction in the metal parts (plenum, flanges, exhaust tube). This residual heat loss $\dot{Q}_{res}$ is shown by the upper field (green) in Fig. 4.23 (right).

The heat flux to the cooling air $\dot{Q}_{duct}$ is relatively constant over the steam dilution ratio,

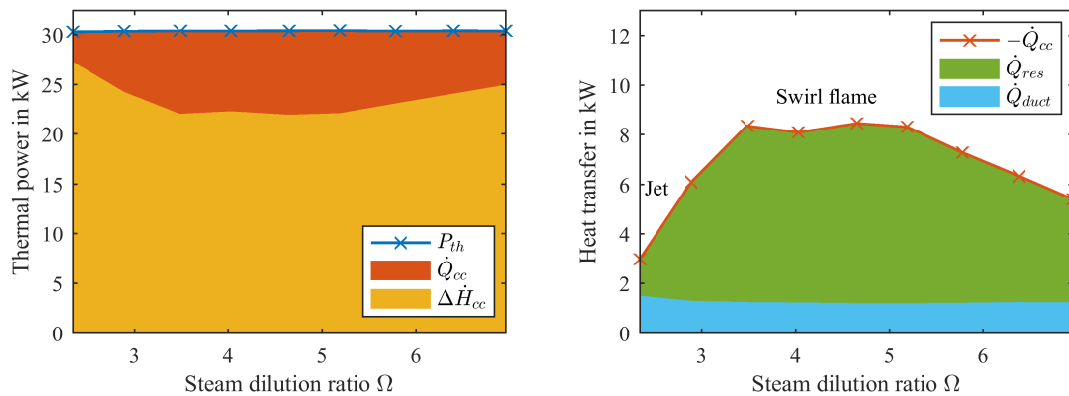


Figure 4.23: Heat balance of Configuration III at 30 kW. Left: Thermal power of the flame and the enthalpy gain of the steam flow. Right: The heat loss of the combustion chamber and the heat input to the cooling air. The green field shows the residual heat loss.

i.e., independent of the flow patterns in the combustion chamber. Consequently, $\dot{Q}_{\text{duct}}$ acts as an offset, and the trends of $\dot{Q}_{\text{cc}}$ and $\dot{Q}_{\text{res}}$ in Fig. 4.23 (right) are very similar. In the jet flame mode, they increase sharply. On the other hand, they stay constant first in the swirl mode and then decline for large Ω . This decline can be explained by the low flame temperature at high Ω , and by the mass flow, which enhances the downstream convection of the heat. Hence less heat is transferred to the metal parts, reducing $\dot{Q}_{\text{res}}$.

The sharp gradient at low steam dilutions is caused by the high exhaust temperature T_{ex} . As stated in the methods section, the T_{ex} is corrected to avoid radiation errors. This correction directly influences the heat loss of the combustion chamber and, thus, the residual heat loss. Figure 4.24 shows that the correction from T_{ex} to $T_{\text{ex,corr}}$ affects mostly the high-temperature range at low steam dilution ratios. Due to the correction, the exhaust temperature gets closer to $T_{\text{ex,ad}}$, which is the ideal exhaust temperature if there would be no residual heat loss. The connection between these temperatures and the corresponding heat transfer is indicated by the face fillings, which have the same color as in Fig. 4.23 (right). It shows that the residual heat loss (green) declines, where the correction increases the exhaust temperature. Without the temperature correction, the residual heat loss would drop not that strong for low steam dilution. The low residual heat loss in this range could indicate that the temperature correction might be too intense for high temperatures. Nevertheless, a correction is necessary due to the significant heat radiation of the thermocouple material in the high-temperature range.

Another reason for the high exhaust temperature at low Ω is that the flame is very confined in the jet mode, leading to a very hot exhaust jet in the center of the combustion chamber. This hot stream hits the thermocouple in the exhaust tube and might cause a small overestimation of the exhaust temperature due to non-uniformity. At moderate steam dilution ratios, the flame transitions to a swirl-stabilized flame and the central recirculation zone establishes in the combustion chamber. Downstream of the IRZ, the temperature profile in the exhaust should be more uniform.

Influence of the Thermal Power and Swirl Intensity

The case, shown in Fig. 4.23 and 4.24, corresponds to a constant thermal power of 30 kW. Varying the flame power leads to the three graphs in Fig. 4.25. The thermal power of the flame only marginally enhances the heat loss of the combustor. At 25 and 30 kW, it first

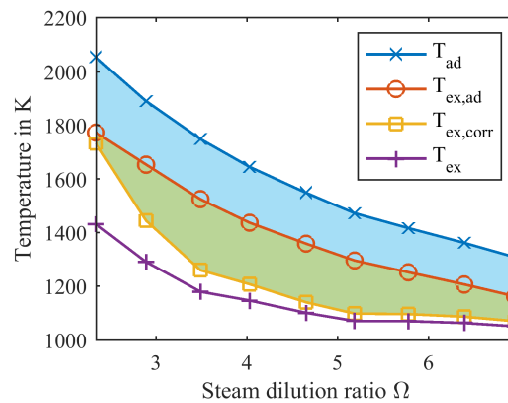


Figure 4.24: Exhaust temperature with and without correction, as well as the adiabatic flame temperature and the ideal exhaust temperature if there would be no residual heat loss.

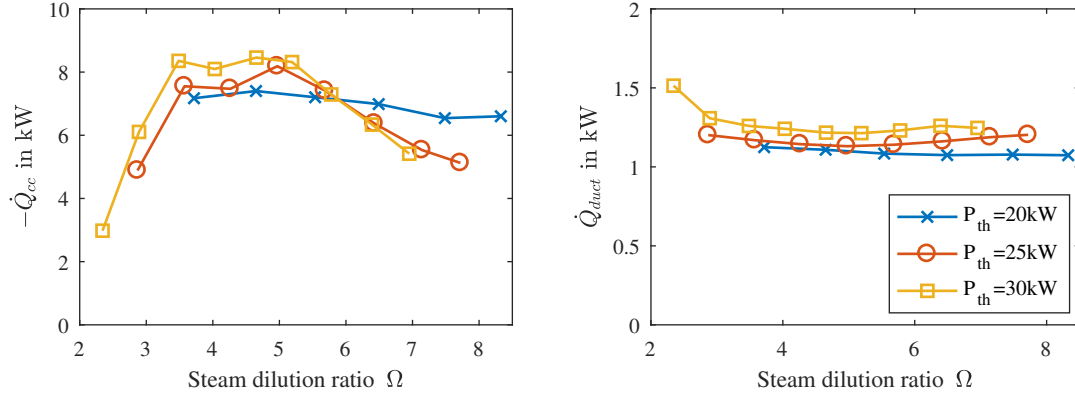


Figure 4.25: Heat flux of Configuration III for three levels of thermal power. Left: Heat loss of the combustor system $\dot{Q}_{cc}$. Right: Heat absorption by the cooling air $\dot{Q}_{duct}$.

increases with Ω and later slightly declines. At 20 kW the heat loss is on a similar level but nearly constant over Ω . According to the right side of Fig. 4.25, the heat transfer to the cooling air $\dot{Q}_{duct}$ is independent of the steam dilution ratio, as observed before, but increases with the thermal power. However, this enhancement is smaller than one would expect by amplifying the thermal power by 25 and 50 %, respectively. The constant cooling flow rate seems to be the largest heat transfer resistance in the system and thus limits the absolute heat transfer from the flame to the cooling air.

Figure 4.26 shows the influence of swirl intensity through the various burner configurations. All configurations show a similar trend as described before for Configuration III but the increased swirl intensity shifts the maximum of the total heat loss $\dot{Q}_{cc}$ towards lower steam dilution ratios. In the case of Configuration XI, the maximum is already at the lowest measured Ω . Here, the flame also does not reach the jet mode because of the high swirl intensity of the steam flow.

The swirl intensity only marginally influences $\dot{Q}_{duct}$. Configuration XI features a little higher cooling power, which might be caused by an increased flame angle and, hence, a closer distance from the flame to the combustion chamber wall (see Fig. 4.19).

In conclusion, the heat flux to the cooling air $\dot{Q}_{duct}$ is nearly independent from the steam dilution ratio, the thermal power and the burner configuration. It indicates that the heat

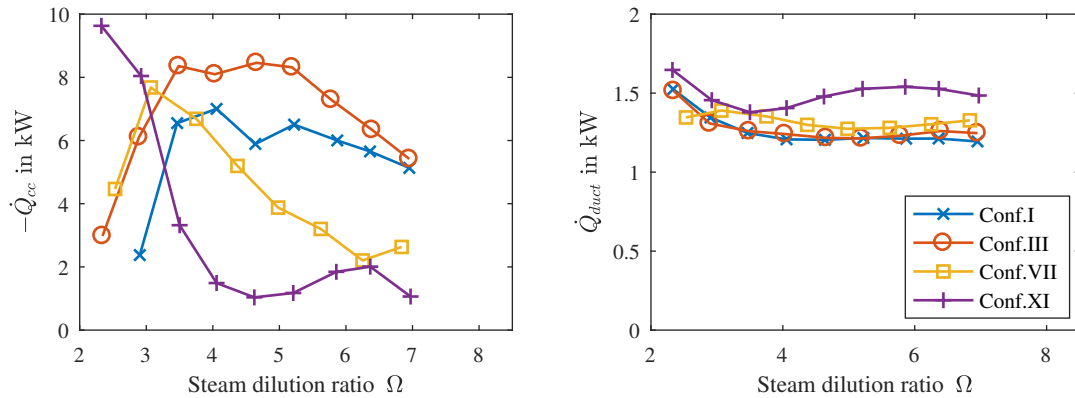


Figure 4.26: Heat flux of different configurations for 30 kW (Configuration VII: 28 kW). Left: Heat loss of the combustor system $\dot{Q}_{cc}$. Right: Heat absorption by the cooling air $\dot{Q}_{duct}$.

flux is already limited by the heat capacity of the cooling air. Hence, water or steam should be used as coolant, if the power density of the combustor is further increased.

4.5.2 Heat Transfer through the Wall

Since the determination of the wall temperatures and Nusselt numbers via the cell method is complex, it is first illustrated for one example flame (Configuration III, 30 kW, $\Omega = 4.7$) along with intermediate values of the network model. Afterwards, the influence of the steam dilution, the thermal power, and the burner configuration on the wall temperatures and the Nusselt numbers is analyzed.

Axial distribution of the heat transfer

The cell method divides the axial coordinate of the combustion chamber into small cells, as shown in Fig. 4.7. Each cell corresponds to a bar in Fig. 4.27 (left). The height of the bars equals the enthalpy difference (Eq. (4.8)) of each cell in the cooling duct. The cells are axially bounded by the positions of the temperature sensors, which are marked by the crosses (x) in the plot. The temperature increases linearly along the length of the cooling duct. Hence, the enthalpy gain is equally distributed over the cells. Only the first cell shows a significantly higher enthalpy gain, because it is located at the flame position, i.e., the point of highest heat release.

The enthalpy gain and the cooling mass flow yield the amount of heat transferred from each combustion chamber cell to the corresponding cooling duct cell. This heat flux $\dot{Q}$ is shown as bars in Fig. 4.27 (right) together with the Nusselt number of each duct cell (Eq. (4.14)-(4.23)). The Nusselt number is nearly constant over x and lies in the order of 80-100. Gnielinski [70] validated the correlation used in this thesis with experimental data for the heat transfer of air in annular ducts [96, 139]. Nasiri et al. [130] and Garimella et al. [66] measured Nusselt numbers in coaxial annular ducts in the order of 60-70 at Reynolds numbers around 10000.

Knowing the Nusselt number and the amount of heat transferred, it is possible to calculate the outer wall temperature of the combustion chamber (Eq. (4.10)-(4.12)). Further, the heat conduction through the wall yields the temperature difference between the inner and outer wall (Eq. (4.24)). Figure 4.28 (left) shows that the inner and the outer wall temperature can

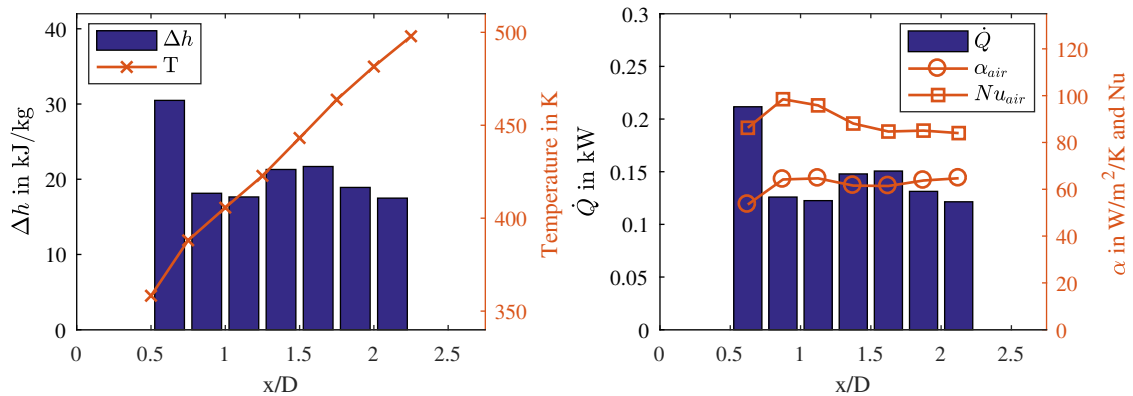


Figure 4.27: Temperature in the cooling duct (left) and Nusselt number in the evaluation cells (right) with the amount of absorbed heat in each cell. Configuration III at 30 kW and $\Omega = 4.7$. The bars refer to the left y-axis, the solid lines to the right one.

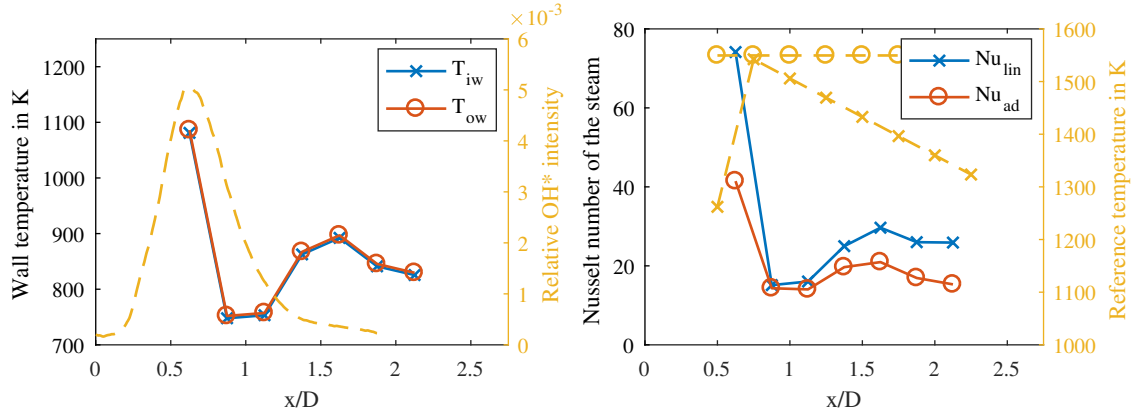


Figure 4.28: Wall temperatures (left) and Nusselt number of the steam in the combustion chamber (right) for two reference temperatures: a linear distribution (x) and the constant T_{ad} (o). Configuration III at 30 kW and $\Omega = 4.7$. The solid lines refer to the left y-axis, the dashed to the right one. The left plot, additionally shows the relative OH* intensity to indicate the flame position.

be considered the same due to the small wall thickness. They are in the range of 700-900 K. Only the first cell shows an extremely high wall temperature of 1100 K due to its increased heat flux. The position of this cell aligns well with the flame position, as the OH* intensity in Fig. 4.28 (left) indicates. Hence, the highest wall temperatures are reached at the point of highest heat release.

As already mentioned in the methods section, the Nusselt number estimation requires a hot gas reference temperature. Figure 4.28 (right) shows two approaches for this reference: the constant adiabatic flame temperature and a linear model, as described in Sec. 4.2.2. Both approaches show qualitatively similar results. Thereby, the linear model yields higher Nusselt numbers, especially at the flame position and in the downstream part of the combustion chamber. Choosing the adiabatic flame temperature as reference results in a nearly constant Nusselt number. As both approaches deliver qualitatively similar axial distributions, only the constant temperature (T_{ad}) approach is used in the following figures. T_{ad} is a well-suited reference because it can be simply calculated from the flame's operating conditions.

The wall temperatures and the Nusselt numbers reveal an interesting axial distribution. The maximum is located close to the flame position, as also reported by Bělohradský et al. [22]. Downstream of the flame however, the wall temperature declines first, before reaching a local maximum at $x/D = 1.5$. The occurrence of the local minimum around $x/D = 1$ might be explained by the strong acceleration of the flow due to the recirculation zone and the thermal expansion. The following local maximum is located at the rear stagnation point of the recirculation zone, where the flow expands and decelerates again. This observation matches the findings from Patil et al. [137], who found the maximum Nusselt number at the point of lowest axial velocity.

Influence of the Steam Dilution Ratio, Thermal Power and Burner Configuration

The observations before refer to a fixed steam dilution ratio of $\Omega = 4.7$. Figure 4.29 shows the influence of the steam dilution on the wall temperature (left) and the Nusselt number (right). The wall temperature decreases with Ω due to the reduced flame temperature. On the contrary, the Nusselt number grows, which is caused by the increased flow rate in the combustion chamber.

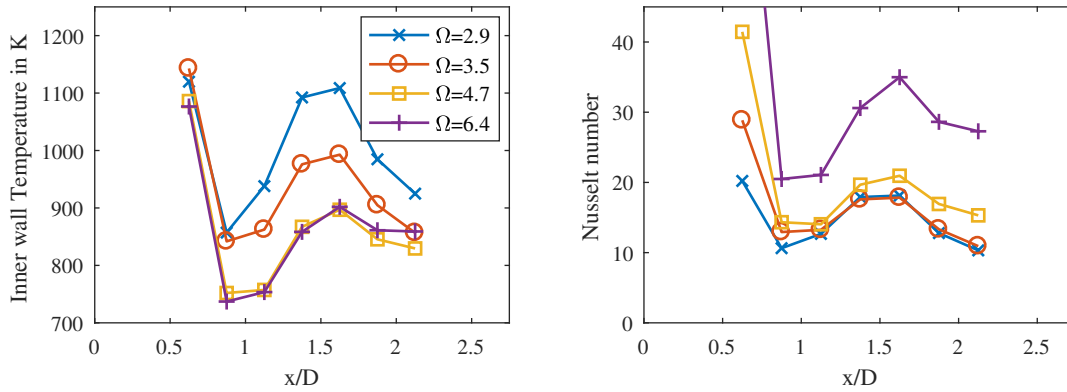


Figure 4.29: Wall temperatures (left) and Nusselt number of the steam in the combustion chamber (right) for different steam dilution ratios Ω . Configuration III at 30 kW with T_{ad} as reference temperature.

The axial distributions of the wall temperature and the Nusselt number are qualitatively similar for all Ω , at each measured thermal power, and for the Configurations I, II, and IV. However, the local maximum does not appear at Configuration III for an unknown reason. Surprisingly, the local maximum's position seems to be independent of the steam dilution ratio, as shown in Fig. 4.29. As stated before, the increasing steam dilution shifts the flame upstream and in the radial direction. It transitions from a jet to a swirl-stabilized flame with a larger spread angle. One could assume that the point of maximum wall temperature moves upstream accordingly. However, the variation of the flame position is only about $0.2D$, which is not resolvable with the current setup since the cells have a width of $0.25D$. Additionally, the total mass flow increases with steam dilution and compensates on the flame position shift due to enhanced convection.

As the axial distribution is almost identical for each steam dilution and thermal power level, the distribution over x/D is reduced to a single number by using the mean wall temperature and the mean Nusselt number. These quantities are shown in Figure 4.30 and 4.31 over the steam dilution ratio for different thermal power levels and burner configurations.

As mentioned before, the wall temperature decreases quickly with the steam dilution due

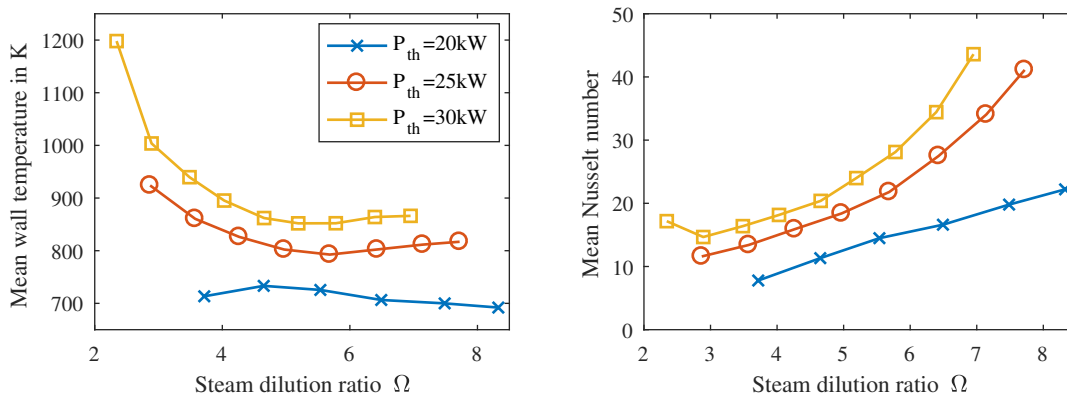


Figure 4.30: Mean wall temperatures (left) and mean Nusselt number of the steam in the combustion chamber (right) over the steam dilution ratio Ω . Configuration III at different power levels with T_{ad} as reference temperature.

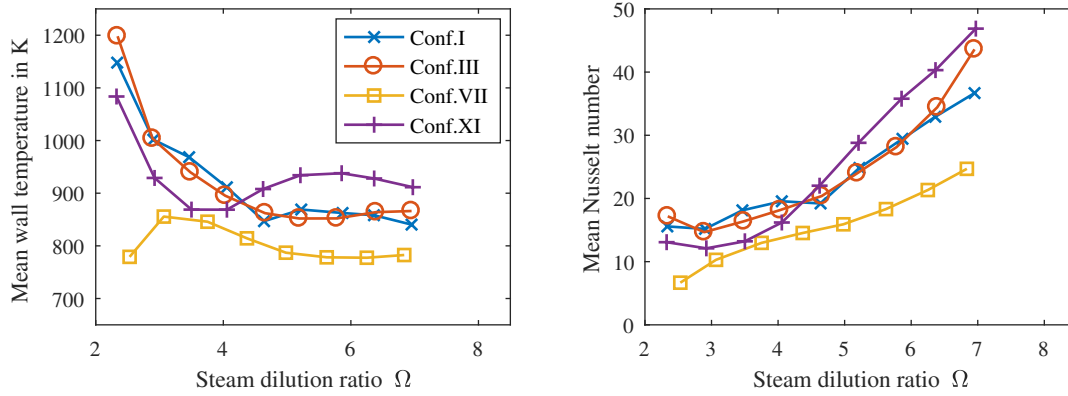


Figure 4.31: Mean wall temperatures (left) and mean Nusselt number of the steam in the combustion chamber (right) over the steam dilution ratio Ω . Various configurations at 30 kW with T_{ad} as reference temperature.

the reduced flame temperature. However, the reduction saturates at $\Omega = 5$. For higher steam dilution ratios, the wall temperature stays constant. As expected, the wall temperature increases with rising thermal power and reaches up to 1200 K at 30 kW. In contrast to the decline at 25 and 30 kW, the wall temperature is constant at 20 kW in the entire observed range.

The Nusselt numbers rise with the thermal power level and with the steam dilution ratio. These effects probably result from the increased flow rates and turbulence in the swirl-stabilized mode.

The burner configuration, on the other hand, has no explicit influence on the wall temperatures and Nusselt numbers, as shown in Fig 4.31. The highest Nusselt numbers are reached with Configuration XI at increased steam dilution. At the same time, Configuration VII shows the lowest Nusselt numbers in this range. Hence, the effect cannot be attributed to increased swirl intensity. However, all configurations provide similar trends over the steam dilution ratio for the wall temperature as well as for the Nusselt number.

Hence general statements can be drawn even though the absolute Nusselt numbers vary between the measurement series. At low steam dilution ratios, the combustion chamber wall has to withstand very high temperatures. Enhanced steam dilution reduces these temperatures to a certain degree, but from $\Omega = 5$ on, there is no further cooling effect to be seen in the data. The Nusselt numbers, on the other hand, grow steadily with the steam dilution.

4.6 Summary

The OH* chemiluminescence images presented two flame types in the combustion chamber: the jet and the swirl-stabilized flame. Which type was present depends on whether VB occurred in the combustion chamber flow or not.

At low steam dilution, a longish jet flame established, which showed the characteristics of a non-premixed flame. Increasing the steam dilution led to the transition of the flame towards a shorter length and a wider spreading angle. Finally, in the swirl-stabilized mode, the flame was located in the SLs of the IRZ.

The operational maps of three burner configurations, shown in this chapter, revealed that the transition ran smoother and occurred at lower steam dilution ratios, the higher the swirl

intensity and thermal power. In terms of adiabatic flame temperatures, the burner was operated in the range of $T_{\text{ad}} = 1000\text{--}2500\text{ K}$. The lower boundary was given by cold blow-off, while the upper limit can be increased by higher thermal power or less dilution. However, it is always restricted by the thermal stress of the combustion chamber.

The temperature measurements of the coaxial cooling flow showed that the amount of heat transferred to the cooling air only increased slightly with the thermal power and the burner's swirl intensity. Moreover, it remained constant over the entire range of steam dilution ratios, even though the dilution significantly reduced the adiabatic flame temperature and altered the flow field. These observations indicated that the heat load of the combustor already exceeded the limits of the air's cooling capacity. In contrast, the total heat loss of the combustion chamber was not constant but featured a maximum for medium steam dilution. The residual heat loss, i.e., the difference between the total heat loss and the heat taken up by the cooling flow, was conducted to the metal parts like the plenum and the flanges.

The axial distribution of the local wall temperatures and Nusselt numbers revealed a local maximum approximately one diameter downstream of the flame. The extremely high wall temperatures, caused by the H_2/O_2 flame, could be reduced by steam dilution down to approximately 900 K at 30 kW. However, the cooling effect saturated at $\Omega = 5$. The Nusselt numbers were in the order of 10-50 and increased with the thermal power and the steam dilution ratio.

This heat transfer study was intended to assist in further burner development for power and pressure scale-up. Already at the lab-scale, the measurements showed that the cooling air captured only a small portion of the combustion chamber heat loss. In more powerful systems, water should be used for cooling because of the higher heat capacity. For instance, the dilution steam could be preheated by the combustor wall before entering the combustor plenum.

CHAPTER 5

Combustion Efficiency Evaluation

The term combustion efficiency in this thesis refers to the avoidance of reactant residuals in the product steam of the H_2/O_2 combustor. The importance of these gas residuals was already highlighted in the first publications of the German H_2/O_2 steam generator (H_2/O_2 -SG) project (see Sec. 1.4.3). Wojkowsky et al. [197] mentioned that residual gas components could arise from deviations of the equivalence ratio from stoichiometric conditions as well as from the incomplete reaction of the reactants. In a cyclic process, these non-condensable gases can increase the condenser pressure, which reduces the cycle efficiency. Besides that, residual gases can damage the components of the power plant. Residual O_2 promotes direct oxidation of the downstream components [20]. H_2 can cause decarbonization of steel components and reduces their lifetime. In combination, both gases form a combustible mixture in the condenser [20]. These important issues allow only a low tolerance of residual gases to apply the H_2/O_2 -SG in power generation cycles successfully. The avoidance of residual gases consists of two primary tasks: Stoichiometric reactant control and high combustion efficiency. The latter will be defined in Eq. (5.10) as the ratio of burned to injected H_2 .

The first section gives a literature review on how the previous H_2/O_2 combustion projects treated this topic. It presents the measurement techniques used, and the residual reactant concentrations targeted and reached in these projects. The latter serves as a benchmark for the current study. Following this review, the experimental setup and the utilized sensors are presented. The results part is divided into measurements with and without equivalence ratio control.

5.1 Literature Review on Combustion Efficiency in H_2/O_2 Combustion

In their H_2/O_2 -SG, the DLR utilized a lambda sensor (LS) to control the flame stoichiometry. Beer et al. [20] described the apparatus and its calibration procedure. They reported that the sensor sensitivity increased significantly for lower concentrations, which is an essential feature for accurate stoichiometry control. The sensor distinguished between H_2 and O_2 residuals and had a very short response time. The DLR measured a sensor reaction time of less than 100 ms, which made it suitable for fast-response control. They also reported that for a low sensor mass flow (265 mg/s), no catalyst was needed in their setup. In the current study, however, a catalyst was necessary, as shown later. Please note that in the cited study, the steam (773 K) and the sensor (1003 K) temperatures were higher than in the experiments of the current thesis.

Fröhlke and Haidn [62] successfully tested the capability of the DLR's H_2/O_2 -SG control system to simultaneously control thermal power, temperature, pressure, and the stoichiomet-

ric ratio of the reactants. For the LS measurements, they sampled exhaust at the end of the vaporization zone. Moreover, they performed gas chromatograph (GC) measurements to determine the H_2 and O_2 concentrations in the exhaust and the combustion efficiency. These tests were performed at 30 MW and 40 bar, which yields a much higher energy density than in the current thesis's lab-scale combustor. The achieved combustion efficiency in their experiments was in the range of 98-99 % [62]. The lowest measured gas residuals were below 0.04 %_m H_2 and 0.3 %_m O_2 . With an optimized injector head for the reactants (and increased thermal power of 58 MW), Haidn et al. [78] were able to show an increased combustion efficiency of more than 99 % but did not succeed in reaching their target window for the residual gas concentrations. Beer et al. [20] reported the target values of 0.3 %_m for O_2 and 0.01 %_m for H_2 . However, these values are process and material-depended. Nevertheless, they will be used as a benchmark in the thesis to give the reader an orientation.

Regarding the combustion efficiency, Beer et al. [20] and Fröhlke and Haidn [62] reported two significant issues in their studies. The first of these being small geometry inaccuracies of the 58 fuel nozzles that led to non-stoichiometric gas mixtures. Moreover, the different thermal loadings of the injectors caused thermal blocking and increased injection inhomogeneities. The residence time in the combustion chamber was only in the order of milliseconds and not long enough to fully mix these local deviations. The second issue at the DLR combustor was the quenching of the reaction. At very high temperatures, the equilibrium composition contains a non-negligible amount of H_2 and O_2 (e.g., 10 %_m O_2 at 4000 K). If the temperature reduction process is much faster than the time scale to set chemical equilibrium, these residuals can partly remain in the low-temperature exhaust gas [77]. In the case of the H_2/O_2 -SG, quenching appeared at the water-cooled walls and in the flue gas due to the sudden temperature reduction by dilution and vaporization of water, injected downstream of the combustion chamber [62].

With the Russian H_2/O_2 -SG, Malysenko et al. targeted a combustion efficiency of 98 % [122] and a concentration of non-condensable gases in the steam of less than 2 %_v [123]. Their evaluation was based on the extraction of steam from the exhaust tube, which subsequently streamed into a condenser. The condensate and the non-condensable fraction drained into a tank, which was ventilated by helium. Afterwards, the gas mixture was analyzed in a GC. For the 25 MW class unit with optimized reactant injectors, they reported a minimum gas concentration in the steam of 0.27 %_v H_2 and 1.07 %_v O_2 . Borzenko et al. [24] reported similar experiments on a 200 kW combustor, which yielded residual hydrogen contents of 0.26-1.21 %_v H_2 . They noticed that increasing the pressure in the combustion chamber improved the combustion efficiency. However, they did not mention any control of the stoichiometry and did not report residual oxygen concentrations. Regarding the measurement technique, they stated that "a certain distortion of the results is possible due to the cooling of the removed steam in the sampler tube" [24], but no uncertainty was given. Additionally, they proposed to estimate the combustion efficiency thermodynamically from the exhaust temperature and pressure, which yielded almost perfect combustion for some configurations.

The design and the working principle of the burner in this thesis (see Sec. 2.3.4) differ significantly from the H_2/O_2 -SG mentioned before. Instead of a high number of individual injectors, the burner features a single central H_2 injection. The residence time is much longer, especially in the swirl-stabilized mode, and the flame temperature is much lower due to the upstream steam dilution. These measures avoid high equilibrium concentrations, which could persist due to quenching. Under the current conditions ($P_{\text{th}} = 30 \text{ kW}$, $T_{\text{in}} = 500 \text{ K}$), the equilibrium concentrations of H_2 and O_2 decrease with Ω for two reasons. The gas residuals are more diluted, and the reduced flame temperature improves the reactant conversion, as

shown in Fig. 5.1. On the other hand, the dilution itself can act as a quenching process due to low reactant concentrations at high steam dilution ratios. In this case, enhanced mixing is necessary (see Sec. 3.4).

The control of the stoichiometric ratio in this thesis is done similarly to the DLR approach by using an LS. A GC evaluation of the combustion efficiency was also intended at the beginning of the project. However, it didn't work out well due to condensation and the broad chromatogram of steam. Instead, a tunable diode laser absorption spectroscopy (TDLAS) O_2 sensor measured the absolute O_2 concentration in the exhaust, enabling the in-situ monitoring of the combustion efficiency. This feature would be beneficial in the industrial application of an H_2/O_2 combustor to continuously observe the system's correct functioning.

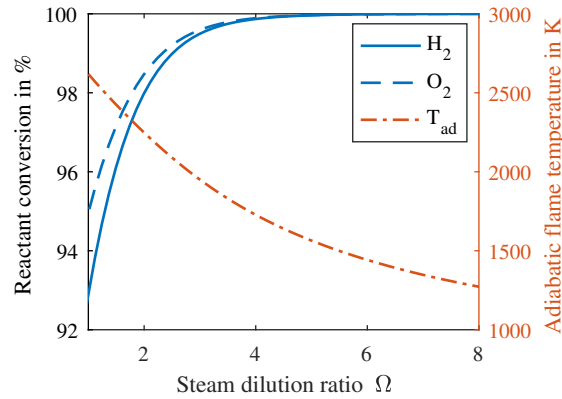


Figure 5.1: Conversion rate of the reactants H_2 and O_2 in the equilibrium state at $P_{th} = 30$ kW and $T_{in} = 500$ K, calculated with the Burke mechanism [31] reaction mechanism.

5.2 Experimental Setup

This section describes the extension of combustion test rig B (CTR B), presented in Sec. 4.1.2, to facilitate automatic control of the equivalence ratio and the measurement of the combustion efficiency. For this purpose, an LS and a TDLAS O_2 sensor are installed in the exhaust segment.

5.2.1 Combustion Test Rig B (Combustion Efficiency)

The combustion tests in this chapter were performed in CTR B, the specifications of which can be found in Tab. 4.1. Figure 5.2 gives a schematic of the working principle of the test rig.

Downstream of the combustion chamber, exhaust gas was extracted by a sampling tube for the LS, which measured deviations of the fuel mixture from stoichiometric conditions (see Sec. 5.3.1). The output voltage of the sensor served as the input for the PID controller, regulating the valve of a bypass O_2 line. If the LS recognize a lean flame, the PID controller reduces the mass flow of the O_2 bypass line. If the flame is fuel-rich it will open the valve. This particular arrangement of a main and a bypass line, as shown in the schematic of Fig. 5.2, had a significant advantage over just one line. The large valve in the mainline was fixed to a mass flow slightly below the desired value, while the PID-operated valve in the bypass line supplied the rest. Hence, the valve's dimensions could be reduced, allowing for a much more precise control of the mass flow. This way, the setup enabled both a substantial variation of the reactant mass flows as well as a fast and accurate control of the mixture

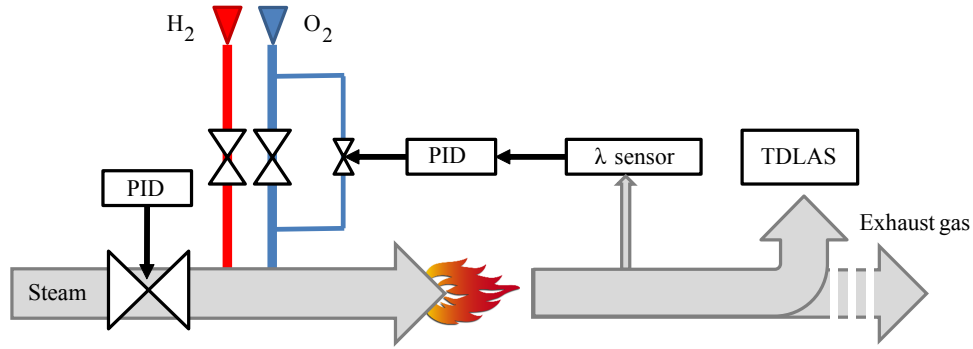


Figure 5.2: Schematic of CTR B with automated equivalence ratio control.

ratio. The gas for the LS was sampled approximately $12.5 D$ downstream of the burner exit. An orifice was placed at the end of the first exhaust tube to homogenize the exhaust stream. Between the sampling tube and the actual LS head, there was a catalyst section, a particle filter, a flow meter, and a manual needle valve. The entire path to the sensor was insulated and externally heated to avoid condensation of the exhaust steam, which could damage the sensor.

The second sensor added to CTR B was a TDLAS system, as described in Sec. 5.3.2. This O_2 sensor is based on the absorption of laser light by O_2 molecules. Since the burner was intended to feature a high combustion efficiency, the expected O_2 concentration in the exhaust was very low. The amount of absorbed light intensity depends on the volume of the laser beam in the measured gas – in other words, on the optical path length L_{op} . In order to still get a clear signal to noise ratio at low concentrations, the optical path length needed to be increased. Since the diameter of the combustion chamber was too small for a cross-sectional measurement, the evaluation took place in a separate measurement section. For this purpose, the flange at the end of the exhaust tubes was blocked by a screen. The exhaust flowed upwards and entered the TDLAS section in the middle, as shown on the right of Fig. 5.3. The exhaust split equally into two parts and convected co-axially to the laser beam in both directions. Symmetric outlet pipes released the exhaust into the atmospheric extraction unit.

The measurement section was insulated to maintain a constant temperature and instru-

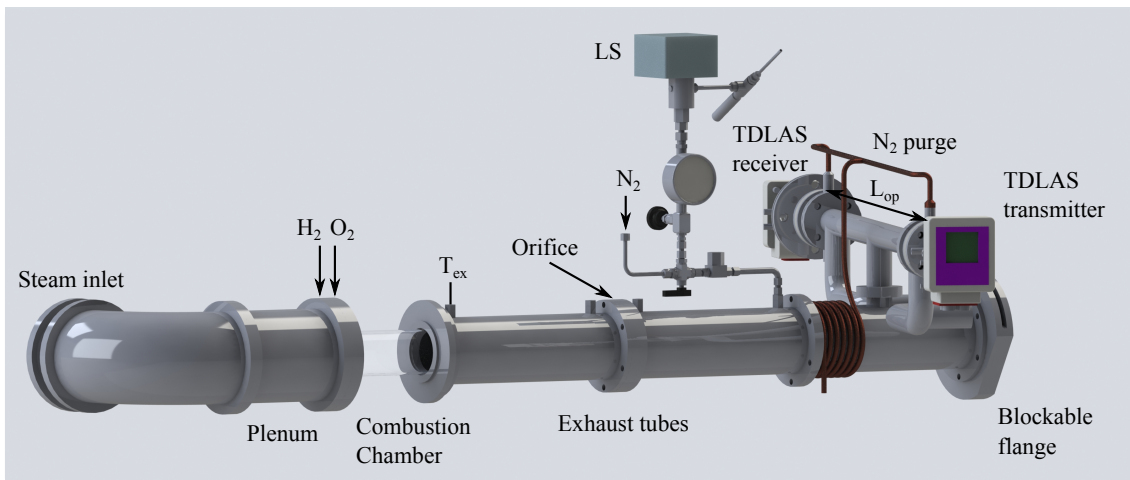


Figure 5.3: Setup of CTR B with LS and TDLAS sensor.

mented with thermocouples and a pressure sensor. Furthermore, the transmitter and receiver modules of the TDLAS system had to be protected from the hot gas. Therefore, cold nitrogen was injected close to the modules as purge. The purge system is further discussed in the following subsection.

5.2.2 Purging of the TDLAS System

The purging of the TDLAS system was intended to protect the transmitter and receiver unit from the hot combustion exhaust, which consisted mainly of steam. At the same time, steam condensation had to be avoided. Therefore, the setup employed a two-staged purge that fulfilled both requirements. Both purge flows, the hot and the cold N_2 purge, can be seen on both sides in Fig. 5.4. Cold purging gas flowed into the measurement tube close to the TDLAS units and cooled down its windows. Additionally, thermal breaks made of ceramic were installed to prevent heat conduction in the metal tube. The N_2 purge flowed through the thermal breaks into the hot gas section. At the junction of the cold purge and the hot steam (union cross), the additional hot N_2 purge was injected. A coil around the exhaust tube preheated the N_2 above the saturation temperature of the steam. This second purge acted as a gaseous thermal barrier to avoid condensation when the steam mixed with the purge gas. All three flows were ventilated through a shared outlet. Only a small portion of the steam flowed into the union cross. The rest was ventilated through the large outlet upstream to safe purging gas. This bypass should not influence the measurement as long as a sufficient portion of steam gets to the union cross so that no purging gas can dilute the exhaust upstream of this point. Accordingly, the distance between both union crosses defined the optical path length of $L_{op} = 1.24$ m. The purging gas was pure nitrogen ($N_2 \geq 99,999\%$, $O_2 \leq 2$ ppm-mol), which does not absorb laser light in the O_2 specific spectral lines and, thus, does not influence the result. In addition to the measurement tube, the TDLAS transmitter and receiver units needed to be purged. In the experiments, the purge flow rates were set using rotameters at standard conditions.

In order to design the purge mass flows properly, a CFD study has been performed. It is presented in Appendix B. The purging flows, which have been chosen from the CFD study, are given in Tab. 5.1.

Table 5.1: Purging flows of the TDLAS system.

	Transmitter	Receiver	
Hot purge	300	300	l/h
Cold purge	400	400	l/h
Unit purge	3	3	l/h

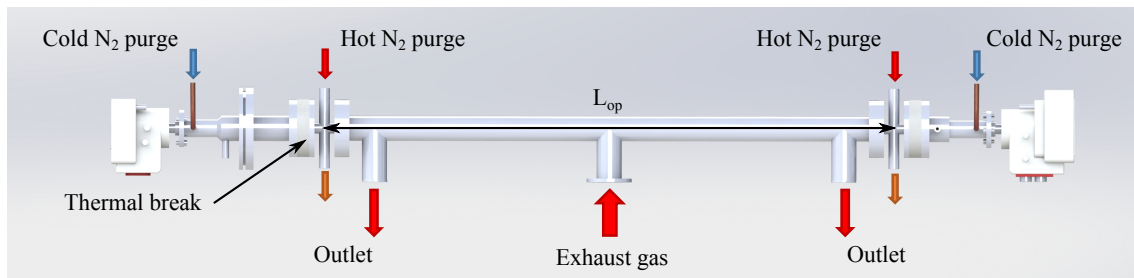


Figure 5.4: Setup of the TDLAS system.

5.3 Methods

The combustion efficiency was evaluated using two sensors, namely the LS and the TDLAS O_2 sensor. Both sensor types are described in detail to understand their output characteristics. Afterwards, the calculation of the combustion efficiency is derived the concentration measurements.

5.3.1 Lambda Sensor

The deployed LS was a ZrO_2 oxygen sensor, which is commonly used to monitor the air-to-fuel ratio in internal combustion engines [28]. The ZrO_2 ceramic is an oxygen-ion-conducting electrolyte that generates a voltage when being in contact with gases of different O_2 concentrations at high temperatures. In the experiments, it was heated up to 1000 K and exposed to the measurement gas on one side, and atmospheric air on the other side of the electrolyte. Assuming complete combustion under perfectly stoichiometric conditions, no O_2 exists in the measurement gas while the reference air has 20.9%_v O_2 . In this case, the sensor generates a threshold voltage. If O_2 is present in the test gas, the lower difference in O_2 concentration between both sides of the sensor leads to a decrease of the voltage signal.

Additionally, the LS is also cross-sensitive to fuel gases. The fuel gas concentration on the measurement side of the sensor leads to a change of the inner material's charge density, which enhances the oxygen-ion-conduction and thus results in an increase of the sensor voltage above the threshold. This behavior allows to measure the concentration of H_2 residual gas in the exhaust gas. However, due to high-temperature pre-oxidation of the stoichiometric parts in the measurement gas, solely the excess component, either H_2 or O_2 , should get in contact with the sensor. Therefore, the LS signal depends only on the deviation from the stoichiometric ratio, not on the total residual gas concentration in the exhaust.

Setup

The LS used in the thesis was a commercial Metrotec A15 sensor. The volume flow through the sensor was fixed to 50 l/h to ensure repeatable measurements and short response times. Only a fraction of the exhaust gas was sampled and analyzed by the sensor. Two ways of sampling the gas for the LS analysis were tested: pressurization of the exhaust gas and suction via a slight vacuum.

In the first case, the probe was installed at the end of the first mixing tube, upstream of an orifice. This position is marked as 1 in Fig. 5.5. The orifice slightly increased the pressure in the exhaust tube. Thus, a small portion of the exhaust gas flowed into the probe, which had three inlets at different axial positions, as shown in Fig. 5.5. The piping to the sensor head was heated and insulated to avoid condensation. Water droplets hitting the electrolyte may evaporate rapidly and lead to sensor damage. Moreover, the LS setup contained a three-way valve for nitrogen purge, a manual flow regulator, a flow meter, and two thermocouples. The first one was located downstream of the probe, the second directly upstream of the LS. Downstream of the sensor head, a tube exhausted the sample to the ambient air.

In the case of sample suction, the overall setup of piping and sensor remained unchanged. However, the sensor was placed downstream of the orifice at approximately $12.5 D$ (marked as 2 in Fig. 5.5). A suction device was attached to the exit of the LS. It consisted of a high-velocity air jet emanating into a smaller chamber and expanding into a diffuser. This generated a slight vacuum inside the sampling probe, sucking sample gas from the exhaust into the LS. The LS was exposed to the same low-pressure as the chamber, which might reduce the O_2 partial pressure on the test gas side and influence the LS voltage

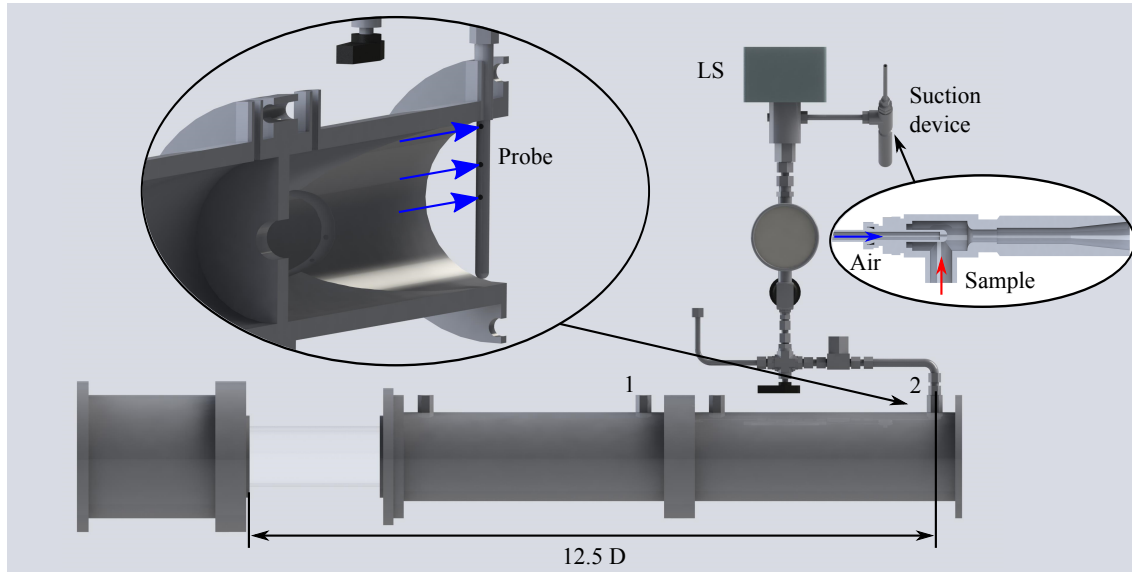


Figure 5.5: Setup of the LS in suction mode.

signal. Therefore, it was crucial to ensure a constant air jet velocity during calibration and measurement.

Calibration and Modeling

A sensor calibration for O_2 and H_2 was conducted with both sampling methods. The calibration was carried out in a simplified test rig without combustor and exhaust tube. One gas, either O_2 or H_2 , was injected far upstream and controlled by a fine metering valve. The pressure difference across the valve was measured and set the test gas mass flow according to Fig. 5.6. The calibration's target was an empirical logarithmic model. Hence, training points were necessary at various gas concentrations to avoid any systematic errors. The mass flow and valve settings were varied during the calibration, as displayed in Tab. 5.2.

Figure 5.7 shows the data points used for the calibration of the LS. The data points were sampled with the suction method, while a second, comparable measurement campaign was conducted for the pressurized sampling method. The O_2 partial pressure difference generated

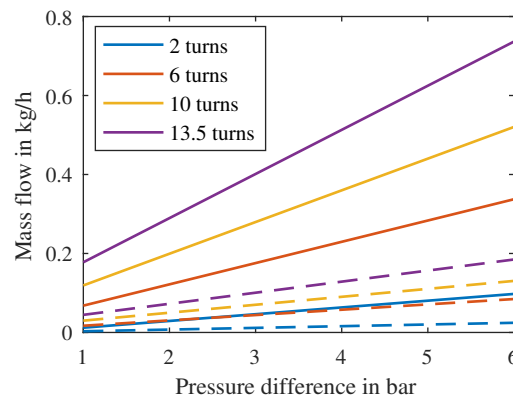


Figure 5.6: Mass flow through the fine metering valve for O_2 (-) and H_2 (- -) depending on the number of valve turns.

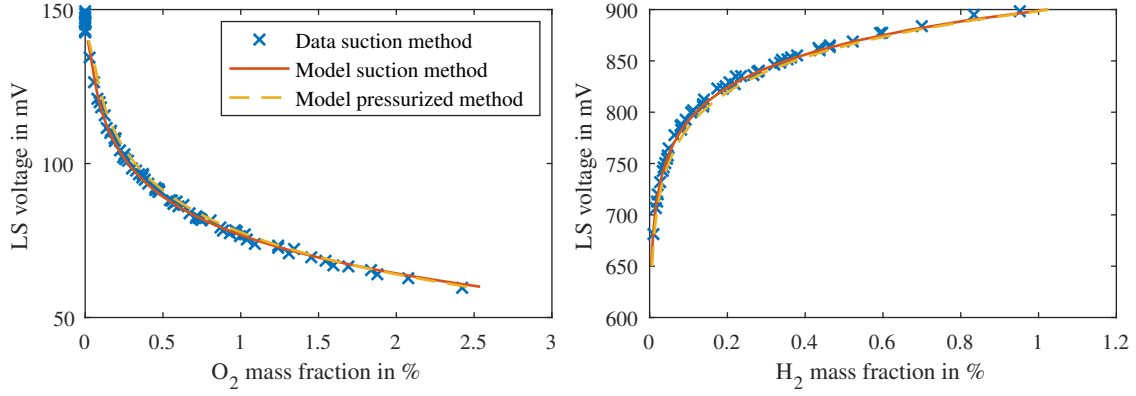


Figure 5.7: Calibration data of the LS in the suction mode with models for both sampling methods.

Table 5.2: LS calibration settings for the suction method.

	O ₂	H ₂	
Valve turns	2, 4, 6, 8, 10, 13.5	2, 4, 8, 13.5	
Steam mass flow	30-50	20-50	kg/h
Gas mass flow	0-0.75	0-0.2	kg/h
Concentration	0-2.5	0-1	% _m

a voltage according to [125]

$$U_{LS} = \frac{RT_{LS}}{4F} \ln \left(\frac{p_1}{p_2} \right), \quad (5.1)$$

where U_{LS} is the electromotive voltage of the LS in V, R is the gas constant in J/mol K, F is the Faraday constant in As/mol, T_{LS} is the temperature in K, p_1 is the O₂ partial pressure on the reference side, and p_2 is the O₂ partial pressure on the measurement side. From this relation, an empirical model for the O₂ mass fraction in the measurement steam was derived. In terms of the test gas mass fraction, the model reads

$$w_{gas} = e^{c_1 \frac{U_{LS}}{T_{LS}} + c_2} \quad (5.2)$$

with c_1 and c_2 as the model constants, which were calculated separately from the calibration data for the suction and the pressurized sampling method, respectively. Figure 5.7 (left) presents the two resulting models for the O₂ mass fraction. Both models nearly collapsed for the whole range of concentrations. Thus, the influence of the sampling mode on the LS voltage was negligible. The only significant difference was that the threshold voltage (pure steam) was reduced in the case of the suction method. The model's deviation from the calibration data in the suction mode was below 0.05 %_m O₂.

The same procedure was applied to model the LS voltage when being exposed to H₂. The empirical model was similar to the O₂ model (Eq. (5.2)) but with an inverted sign of the constant c_1 . Figure 5.7 (right) shows that also for H₂ both models were in good agreement. The model's deviation from the calibration data in the suction mode was below 0.01 %_m H₂.

In order to use the LS as input for a PID controller, the models for O₂ and H₂ needed to be combined. A threshold voltage was defined, at which the model coefficients switch from O₂ to H₂. If the flame burns perfectly stoichiometric, the LS should generate the threshold

voltage, corresponding to the equivalence ratio $\phi = 1$. If the flame deviates towards a lean equivalence ratio, the voltage decreases under the threshold, and vice versa for rich mixtures. The equivalence ratio can be determined from the calculated mass fractions in Eq. (5.2). The relation between ϕ and the LS voltage is shown in Fig. 5.8 for three levels of steam dilution. At $\phi = 1$, which was the desired operating point of the flame, there is a huge jump in the LS voltage. The LS can, therefore, clearly distinguish between lean and rich deviations. During the measurements, the PID controller regulated the valve such that the LS voltage remains close to the jumping point, i.e., at the threshold voltage, which was approximately $U_{\text{threshold}} \approx 140 \text{ mV}$ in the experiments.

As Fig. 5.8 points out, the LS voltage has a logarithmic behavior (see Eq. (5.1)) and is, thus, not well suited as PID input. Using the empirical LS model, a generic PID input function is defined, which reads

$$w_{\text{control}} = \begin{cases} e^{c_{1,\text{O}_2} \frac{U_{\text{LS}}}{T_{\text{LS}}} + c_{2,\text{O}_2}} & , \quad U_{\text{LS}} < U_{\text{threshold}} \\ 0 & , \quad U_{\text{LS}} = U_{\text{threshold}} \\ -s \cdot e^{c_{1,\text{H}_2} \frac{U_{\text{LS}}}{T_{\text{LS}}} + c_{2,\text{H}_2}} & , \quad U_{\text{LS}} > U_{\text{threshold}} \end{cases} \quad (5.3)$$

The H_2 mass fraction got a negative sign to differentiate between both gases. Moreover, it was multiplied by the stoichiometry factor s because there is nearly eight times more oxygen in a stoichiometric flame. Without this factor, lean deviations would have a much larger effect on the PID than rich deviations.

The control mechanism was not perfect and slight fluctuations of the equivalence ratio around stoichiometric were typically still observed. However, these deviations can be quantified using the LS signal. Here, the same models were applied to calculate the mass fraction of the excess gas. For the O_2 mass fraction, it was extended by a linear blending function to avoid a jump to zero at the threshold voltage.

$$w_{\text{O}_2, \phi} = \begin{cases} e^{c_{1,\text{O}_2} \frac{U_{\text{LS}}}{T_{\text{LS}}} + c_{2,\text{O}_2}} & , \quad U_{\text{LS}} < 130 \\ \frac{e^{c_{1,\text{O}_2} \frac{130 \text{ mV}}{T_{\text{LS}}} + c_{2,\text{O}_2}}}{15 \text{ mV}} & , \quad 130 \leq U_{\text{LS}} \leq 145 \\ 0 & , \quad U_{\text{LS}} > 145 \end{cases} \quad (5.4)$$

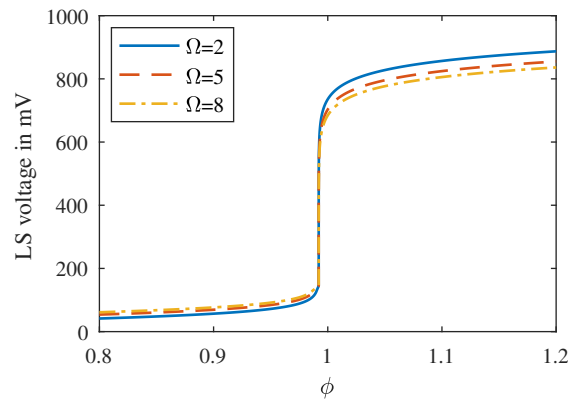


Figure 5.8: Modelled LS voltage over the equivalence ratio of a 30 kW flame for three levels of steam dilution Ω .

$$w_{\text{H}_2, \phi} = \begin{cases} 0 & , \quad U_{\text{LS}} \leq 200 \\ e^{c_{1, \text{H}_2} \frac{U_{\text{LS}}}{T_{\text{LS}}} + c_{2, \text{H}_2}} & , \quad U_{\text{LS}} > 200 \end{cases} \quad (5.5)$$

All voltage limits on the right hand side of the equation are given in millivolts. The mass fractions $w_{\text{O}_2, \phi}$ and $w_{\text{H}_2, \phi}$ were important factors for the calculation of the combustion efficiency as described in Sec. 5.3.3.

Catalyst

In initial combustion tests, the LS indicated stoichiometric conditions. Simultaneously, the mass flows of the reactants were measured by Coriolis mass meters, which allowed a second calculation of the equivalence ratio. In some operating points, the system worked properly, and both measurements showed $\phi \approx 1$. However, especially at the edges of the operational range, the Coriolis measurement showed a significant deviation towards lean conditions ($\phi_{\text{MM}} \leq 1$). In this case, the membrane of the LS should get in contact with the excess O_2 and generate a voltage lower than the threshold, i.e., ϕ_{LS} should be lean instead of stoichiometric. This behavior indicated a malfunction of the LS measurement and can have two reasons.

First, the sample taken from the exhaust gas may not be representative, which means that the mixing length downstream of the flame was too short to generate a homogeneous exhaust stream. Tests with different sample positions and methods (pressurized, suction) indicated that this is unlikely. In the case of the suction method (see Fig. 5.5), the exhaust gas passed an orifice before entering the LS probe at three radial positions. Hence, the exhaust gas was considered fully mixed. Nevertheless, the error still appeared.

A malfunction of the LS itself was most probably caused by incomplete pre-oxidation. This would mean that the sensor membrane got in contact with H_2 and O_2 at the same time. In this case, the LS would be more sensitive to H_2 and would have measured a voltage above the threshold [20]. Hence, it signaled that the flame was rich even if it was stoichiometric. The LS was heated up to approximately 1000 K, which should lead to a pre-oxidation of both gases before they reached the sensor surface. This well-proven method for air-based exhaust streams seemed to fail for the steam flows in the current study.

This hypothesis was verified in an isothermal test, where small amounts of H_2 and O_2 were added to a constant flow of superheated steam. Thereby, the O_2 mass flow was conserved at a fixed mass flow, while the H_2 mass flow was increased by the fine metering valve from zero to rich conditions. The mixture was not ignited but directly delivered to the LS. In Fig. 5.9 (left) the LS voltage of this test is compared to the theoretical value of the model. It shows that the LS jumped much too early (already at $\phi_{\text{LS}} = 0.5$) from lean to rich. This behavior confirms the hypothesis that the H_2 was not fully converted by pre-oxidation. The pre-oxidation may have been quenched by the high amount of steam in the exhaust, which exceeded 99 %.

This observation led to the conclusion that some kind of catalytic material was needed to assist the pre-oxidation of the stoichiometric residual gas components. A well-proven catalytic material for H_2 [88] and hydrocarbons [80] is platinum. Figure 5.10 shows the two implemented catalyst concepts. In the concept of Catalyst A, a 2 g wire of pure platinum was placed in the 6 mm tubing. The wire had a diameter of 0.3 mm and a total length of 1.3 m. It was wrapped to a length of 0.1 m to increase the probability that the reactants get in contact with the catalyst. In the concept of Catalyst B, on the other hand, the

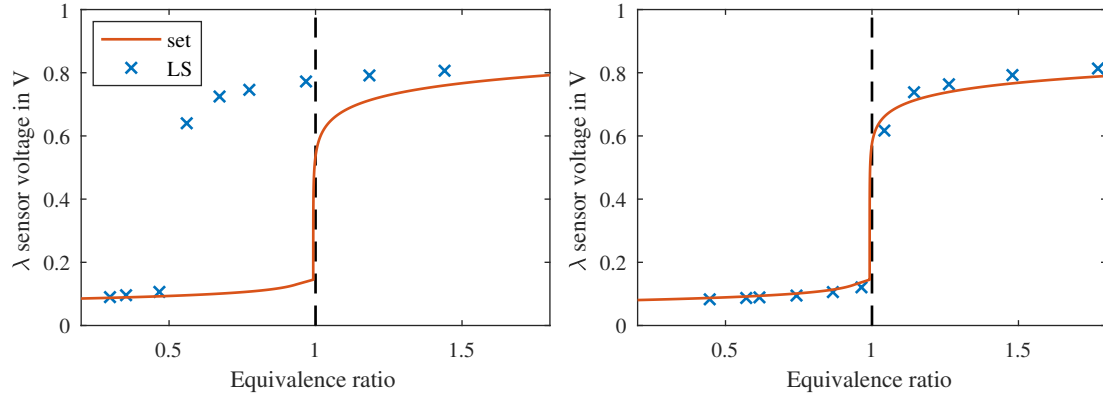


Figure 5.9: LS voltage under exposure of a steam/ H_2 / O_2 mixture without catalyst (left) and with Catalyst A (right). Both conducted without flame and H_2 measured with the fine metering valve.

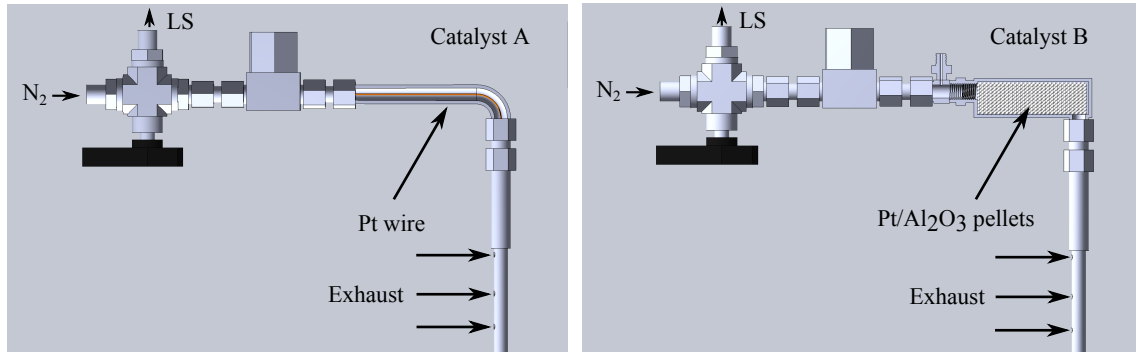


Figure 5.10: Catalyst A (left) and B (right) installed in the pipe between exhaust probe and LS.

platinum was not pure but supported by alumina pellets. This option is more cost effective and significantly increases the surface of the catalytic material. The pellets were filled into a box of 68x20x20 mm. The gas entered from the bottom and flowed through the whole box. This way, also the residence time was increased, enabling a sufficient pre-oxidation. Haneda et al. [80] showed that the conversion rate of hydrocarbons using alumina-supported platinum reaches 100 % for temperatures above 600 K. Therefore, the catalyst was heated up by external heating to approximately 800-830 K in the experiments.

Figure 5.9 (right) shows the results of the validation test with a steam/ H_2 / O_2 mixture using Catalyst A. The good agreement between the measured values and the theoretical curve prove the successful pre-oxidation with the catalyst. The same result was achieved with Catalyst B. However, it reduced the response time of the LS and was therefore not suitable for the combustion tests. Furthermore, the catalyst box accumulated hydrogen when being temporarily operated at rich conditions. For these reasons, Catalyst A was used for the combustion efficiency measurements.

5.3.2 Tunable Diode Laser Absorption Spectroscopy

A TDLAS system acted as an in-situ O_2 sensor to quantify the combustion efficiency and the residual gas concentration in the exhaust. In general, TDLAS systems make use of the concentration-dependent line shape variations through light absorption. Absorption lines

appear at individual wavelengths for each species. They are well suited for measurements in harsh environments. Example applications were reported in fire suppression tests with large fractions of water (vapor and drops) [13, 163] and the high-temperature exhaust of methane combustion in an O_2/CO_2 atmosphere [35]. In the particular case of the H_2/O_2 burner, only minimal concentrations of O_2 were expected in the exhaust gas. However, the commonly used direct TDLAS method is less applicable at low gas concentrations due to low signal-to-noise ratios [167]. Instead, the wavelength modulation spectroscopy (WMS) technique was applied, which modulates the laser wavelength sinusoidally around the wavelength of the absorption line [97]. Here, the concentration is derived from the amplitude of the second harmonic of the modulation frequency. A theoretical overview to this technique was given by Rieker et al. [150]. Neethu et al. [131] validated the applicability of this technique for O_2 concentration measurements. The accuracy and the lower detection limit of such sensors depend mainly on the optical path length. Since the evaluation of the combustion efficiency requires high accuracy and a low detection limit, the laser beam was not directed perpendicular through the exhaust tube, which would only lead to an optical pathway of $l = 0.1$ m. Instead, the exhaust was guided into a measurement tube parallel to the laser beam, as already described in Sec. 5.2.

The TDLAS system delivered the volume fraction v_{O_2} of O_2 in the measurement section. It was subsequently converted into the O_2 mass fraction of the exhaust by

$$w_{\text{O}_2,\text{ex}} = \frac{\rho_{\text{O}_2} v_{\text{O}_2}}{\rho_{\text{O}_2} v_{\text{O}_2} + \rho_{\text{H}_2\text{O}} (1 - v_{\text{O}_2})}. \quad (5.6)$$

Thereby, the densities were determined by the temperature in the measurement section, which was gathered by a thermocouple.

Calibration and Validation

The signal of the TDLAS sensor was sampled at $f_s = 1$ Hz and time-averaged over $t = 180$ s during the calibration. Additionally, the transmission of the laser was measured to ensure correct measurements. Even at the harsh conditions of the high-temperature exhaust, the transmission of the laser was always larger than 70 %. At calibration conditions, the transmission was between 95–100 %.

Under standard conditions (N_2 , 293.25 K, $L_{\text{op}} = 0.512$ m), the system showed a deviation from linearity of less than 0.1 %_v O_2 , but for the actual conditions in high-temperature steam, the TDLAS system required a new calibration. The calibration was conducted in CTR B without a flame or H_2 injection. Since the output signal of the TDLAS sensor is linearly dependent on the volumetric O_2 concentration, it only required a two-point calibration. The selected points were 0 and 2.9 %_v O_2 . The steam temperature during the calibration was lower than the exhaust temperature expected during the combustion experiments. Therefore, the concentration measurement was corrected by the temperature in the TDLAS section. The correction is integrated into the commercial system and was validated by its manufacturer. Figure 5.11 (left) shows that the sensor output is not significantly influenced by the steam temperature at fixed O_2 concentration.

On the other hand, the variation of the mass flow in Fig. 5.11 (right) showed a linear dependency. The measured signal was slightly below the set value if the steam mass flow rate was lower than at calibration conditions (50 kg/h). This deviation was caused by the purging concept (Sec. 5.2.2). As there has been no window to separate the purge gas and the steam, a variation of the steam mass flow could cause little changes in the O_2 -containing laser volume, i.e., the optical path length changes. The measured concentration was a linear

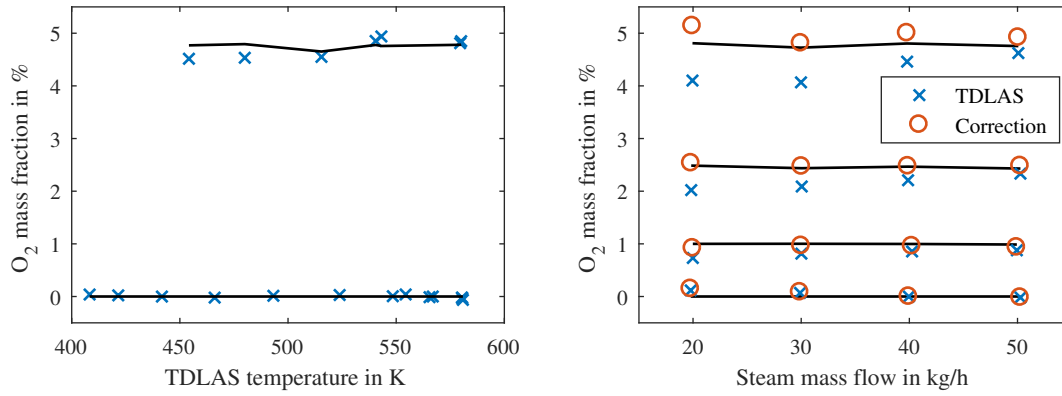


Figure 5.11: Influence of temperature (left) and steam mass flow (right) on the TDLAS-measured O₂ concentration. The lines represent the preset concentration, while the stars are the TDLAS values and the circles the mass flow corrected values.

function of this path length, and so the influence of the mass flow is linear as well. Since it was not handy to change the optical path length during the measurement to adjust to different mass flows, the concentration was corrected afterwards by a linear function. The circles in Fig. 5.11 show the corrected values, which coincided with the set values (solid line).

Subsequently to the calibration, the measurement of the TDLAS sensor and the LS were validated simultaneously by a steam flow with different concentrations of O₂. The conditions of the measurement were varied according to Tab. 5.3. Thereby, the temperature refers to the temperature in the TDLAS measurement section. It was varied at a constant steam mass flow of 50 kg/h. On the other hand, the variation of the steam mass flow also caused a variation of the temperature due to increased heat loss. While at 50 kg/h, the temperature reached 573 K, it dropped to 473 K at 20 kg/h.

Table 5.3: TDLAS validation settings.

	O ₂	
Steam mass flow	20, 30, 40, 50	kg/h
Temperature	473-573	K
Gas mass flow	0-0.75	kg/h
Concentration	0-2.7	% _m

Since the steam in the validation test contained only O₂, the LS should measure the same values as the TDLAS O₂ sensor. The measurement points of both devices were taken simultaneously and are plotted in Fig. 5.12 against the preset O₂ mass fraction. The data points show a standard deviation from the set value of $\sigma_{LS} = 0.07\%_m$ and $\sigma_{TDLAS} = 0.09\%_m$.

5.3.3 Combustion Efficiency Calculation

As noted before, reactant residuals must be strictly limited in the exhaust gas. These residuals can appear due to combustion inefficiencies in the flame and due to deviations of the equivalence ratio. Both factors must be accounted for in the measurements.

First, the deviation of the stoichiometric H₂/O₂ ratio was determined by the LS using the model in Eq. (5.4) and (5.5). This model yields either $w_{H_2,\phi}$ or $w_{O_2,\phi}$, depending on the excess concentration of both gases. Since the flame is either rich or lean, only one of both can be non-zero at the same time. However, in the measurements, the LS signal was

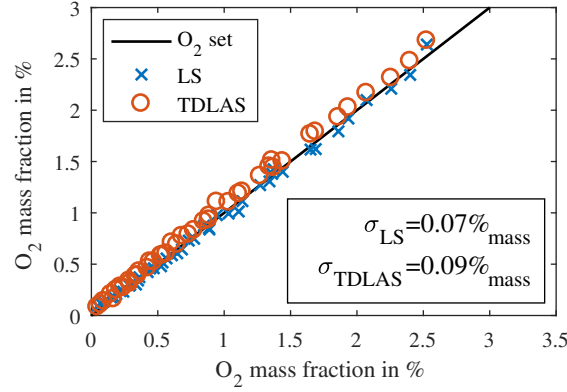


Figure 5.12: Simultaneous validation measurements of the TDLAS and LS.

recorded for 180s and time-averaged subsequently. Fluctuations of the equivalence ratio can trigger a change from rich to lean or vice versa within the measurement time. Hence, after time-averaging, both quantities can be non-zero for the same measurement point. Besides the measurement of the non-stoichiometric concentrations, the LS also controlled the equivalence ratio and balanced out deviations via a PID controller.

Since in a perfectly stoichiometric flame both $w_{H_2,\phi}$ and $w_{O_2,\phi}$ would be zero, the absolute O_2 concentration in the exhaust $w_{O_2,ex}$ is required to evaluate combustion inefficiencies. For this task, the TDLAS sensor (see Sec. 5.3.2) was measured $w_{O_2,ex}$ in addition to the signals recorded by the LS.

Typically, the combustion efficiency characterizes the amount of fuel that is converted in the flame. In the current thesis, it was the ratio of burned H_2 to the total H_2 mass flow. Therefore, the concentration of unburned O_2 in the exhaust was used to estimate the amount of burned H_2 . The non-stoichiometric O_2 concentration $w_{O_2,\phi}$ was subtracted from the O_2 in the exhaust to compensate for deviations of the equivalence ratio. Hence, the unburned mass fraction of O_2 was

$$w_{O_2,ub} = w_{O_2,ex} - w_{O_2,\phi}. \quad (5.7)$$

Since this fraction was solely the stoichiometric part of the O_2 residual in the exhaust, the mass fraction of the unburned H_2

$$w_{H_2,ub} = \frac{1}{s} w_{O_2,ub} \quad (5.8)$$

could be derived by multiplication with the stoichiometric mass ratio s , which is defined by the molar masses of the reactants as

$$s = \left(\frac{\dot{m}_{O_2}}{\dot{m}_{H_2}} \right)_{st} = \frac{1}{2} \frac{M_{O_2}}{M_{H_2}} = 7.9367. \quad (5.9)$$

The multiplication of this mass fraction with the mass flow rate of the exhaust $\dot{m}_{ex} = \dot{m}_{ct} + \dot{m}_{H_2} + \dot{m}_{O_2}$ yielded the mass of unburned H_2 in the exhaust. The combustion efficiency was then defined to

$$\eta_c = \frac{\dot{m}_{H_2} - w_{H_2,ub} \dot{m}_{ex}}{\dot{m}_{H_2}}. \quad (5.10)$$

As additional important information, the total H_2 concentration in the exhaust could be

estimated by

$$w_{\text{H}_2,\text{ex}} = w_{\text{H}_2,\text{ub}} + w_{\text{H}_2,\phi}. \quad (5.11)$$

Now, if there is no LS available for stoichiometry control, the combustion efficiency can still be calculated by using the mass meters of the fuel supply system. However, there would be no automatic control for the equivalence ratio of the flame in this case. Instead, the non-stoichiometric mass fraction had to be calculated from the Coriolis mass meter readings $\dot{m}_{\text{H}_2}$ and $\dot{m}_{\text{O}_2}$. Since these mass flows were time-averaged values, they could only yield one non-stoichiometric component, either $w_{\text{H}_2,\phi}^*$ or $w_{\text{O}_2,\phi}^*$, which were distinguished by the sign of the resulting value.

$$w_{\text{O}_2,\phi}^* = \begin{cases} \frac{\dot{m}_{\text{O}_2} - s\dot{m}_{\text{H}_2}}{\dot{m}_{\text{ex}}} & , \quad \dot{m}_{\text{O}_2} - s\dot{m}_{\text{H}_2} > 0 \\ 0 & , \quad \text{else} \end{cases} \quad (5.12)$$

$$w_{\text{H}_2,\phi}^* = \begin{cases} -\frac{\dot{m}_{\text{O}_2} - s\dot{m}_{\text{H}_2}}{\dot{m}_{\text{ex}}} & , \quad \dot{m}_{\text{O}_2} - s\dot{m}_{\text{H}_2} < 0 \\ 0 & , \quad \text{else} \end{cases} \quad (5.13)$$

Further, the mass fractions of unburned reactants are

$$w_{\text{O}_2,\text{ub}}^* = w_{\text{O}_2,\text{ex}} - w_{\text{O}_2,\phi}^* \quad (5.14)$$

and

$$w_{\text{H}_2,\text{ub}}^* = \frac{1}{s} w_{\text{O}_2,\text{ub}}^*. \quad (5.15)$$

As this yielded the same quantities as before, the formulation of the combustion efficiency and the total H_2 fraction in the exhaust did not change.

$$\eta_c^* = \frac{\dot{m}_{\text{H}_2} - w_{\text{H}_2,\text{ub}}^* \dot{m}_{\text{ex}}}{\dot{m}_{\text{H}_2}} \quad (5.16)$$

$$w_{\text{H}_2,\text{ex}}^* = w_{\text{H}_2,\text{ub}}^* + w_{\text{H}_2,\phi}^* \quad (5.17)$$

The main difference between both approaches was that the first yielded time-resolved mass fractions of the non-stoichiometric reactant species. Hence equivalence ratio fluctuations were recorded and influenced the combustion efficiency. In the second approach, these fractions were calculated from the time-averaged reactant mass flow rates. Thus, the temporal equivalence ratio fluctuations were neglected. If these fluctuations were around $\phi = 1$, for instance, they could have canceled out the deviations from stoichiometry completely, leading to $w_{\text{O}_2,\text{ub}} = w_{\text{O}_2,\text{ex}}$ and $w_{\text{H}_2,\text{ub}} = w_{\text{H}_2,\text{ex}}$. However, by neglecting the temporal fluctuations of $w_{\text{O}_2,\phi}$, the combustion efficiency might decrease, i.e., $\eta_c^* \leq \eta_c$. Hence, η_c^* should be a rather conservative estimation of the combustion efficiency. Yet, there was a lag of accuracy when the equivalence ratio was set according to the mass flow meter readings. This issue is discussed in the next two sections.

5.4 TDLAS-based Combustion Efficiency Measurements

The following two sections present the results of the combustion efficiency measurements of the oxyfuel hydrogen burner. The first one refers to the combustion efficiency η_c^* , as defined in Eq. (5.16). It is based solely on the TDLAS oxygen sensor measurements. The equivalence ratio was not controlled but set to stoichiometric conditions according to the mass meter readings. In each measurement series, the steam dilution was varied, while keeping the reactant mass flow rates as constant as possible.

5.4.1 Variation of thermal power

First, the thermal power level was varied for a single burner configuration. The TDLAS sensor was used to measure the total O_2 concentration in the exhaust, which is shown in terms of the mass fraction in Fig. 5.13 (left). It reveals that increasing the thermal power level reduces the O_2 residuals. For the lowest dilution ratios, the combustion takes place in the jet flame shape. In this regime, the 26 and 31 kW flames show residual O_2 concentrations below 0.3 %_m, which was the targeted concentration in the DLR project [20]. This benchmark is given by the dashed lines in Fig. 5.13 as an orientation for the reader. The 30 kW flame features slightly higher gas residuals in the transition phase. When reaching the swirl-stabilized shape, $w_{O_2,ex}$ reduces again to 0.2 %_m and stays low up to a steam dilution ratio of $\Omega = 6$. At even higher dilution ratios, the residual concentration increases quickly as the flame approaches the blow-off limit.

As mentioned before, the measured concentration in the exhaust splits into two parts: unburned O_2 due to combustion inefficiencies and excess O_2 if the flame is lean instead of stoichiometric. In order to calculate the H_2 concentration, both parts must be known. The excess concentrations, $w_{O_2,phi}^*$ and $w_{H_2,phi}^*$, were calculated by Eq. (5.12) and (5.13). The equivalence ratio was determined from the time-averaged reactant mass flow rates. Hence, it is a time-averaged quantity itself and can only yield one excess gas concentration of either H_2 or O_2 . Consequently, the O_2 concentrations in Fig. 5.14 are zero for almost every measurement point because the flame is slightly rich according to the Coriolis mass meters. The excess concentrations are much lower than the gas concentrations in the exhaust. Thus, most of the residual gas is the result of combustion inefficiency. Consequently, the trend of the unburned H_2 in Fig. 5.13 is nearly the same as seen for the H_2 fraction in the exhaust, i.e., $w_{H_2,ub} \approx w_{H_2,ex}$.

The combustion efficiency was calculated from $w_{H_2,ub}$ by applying Eq. (5.17). It is plotted in Fig. 5.15 (right) over the steam dilution ratios for three levels of thermal power. The jet flame at the lowest steam dilution reveals a very high combustion efficiency of up to 99.25 %. While going through the flame type transition, the efficiency slightly decreases, before reaching a local maximum in the swirl-stabilized mode at 30 kW. The increase of the steam dilution triggers two contradicting effects. On the one hand, it causes a reduction of the reactant concentration, which is expected to impair the combustion efficiency because local zones of non-stoichiometric reaction can appear. On the other hand, the recirculation zone in the swirl-stabilized mode enhances the mixing and preheating of the reactants, and may further lead to a recirculation of unburned gas to the reaction zone. These advantages dominate up to a dilution ratio of $\Omega = 5.5$ in the 30 kW case and increase the combustion efficiency. For further increased steam dilution, the effects of the reduced reactant concentration and the very low flame temperature are dominant. Hence the combustion efficiency drops at high Ω .

For lower thermal power, the local maximum is not visible because the transition to the swirl-stabilized mode takes place at higher steam dilution ratios, as described in Sec. 4.4.

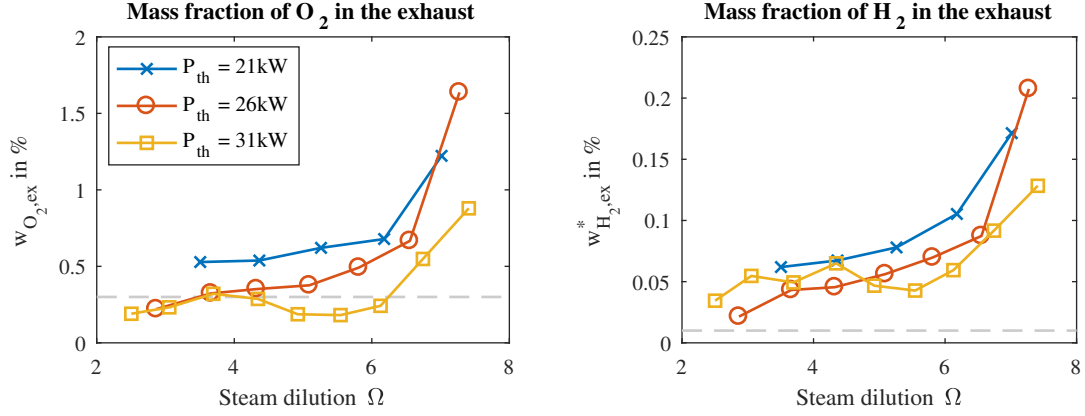


Figure 5.13: Residual gas mass fractions in the exhaust for burner Configuration III. Left: O₂ measured by the TDLAS sensor. Right: H₂ calculated using Eq. (5.17). The dashed lines mark the target concentrations in the DLR project [20].

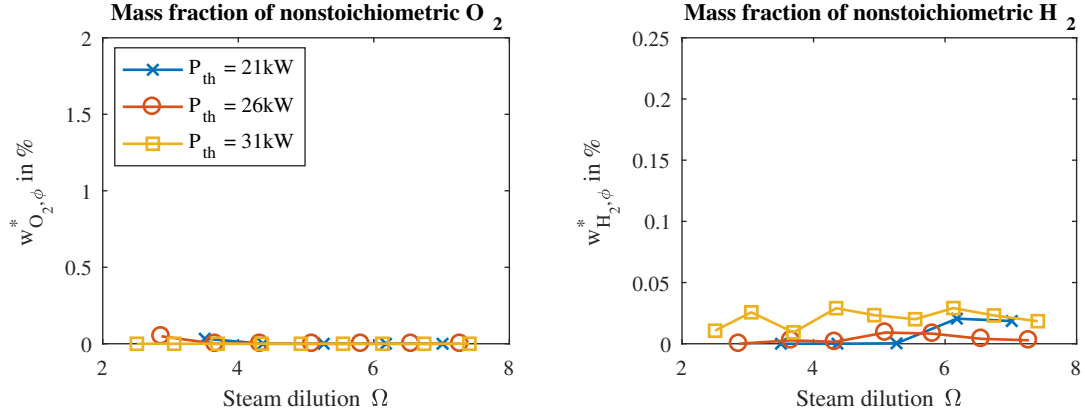


Figure 5.14: Deviation from the stoichiometric ratio for burner Configuration III. Left: O₂ measured by the Coriolis mass meters and Eq. (5.12). Right: H₂ measured by the Coriolis mass meters and Eq. (5.13).

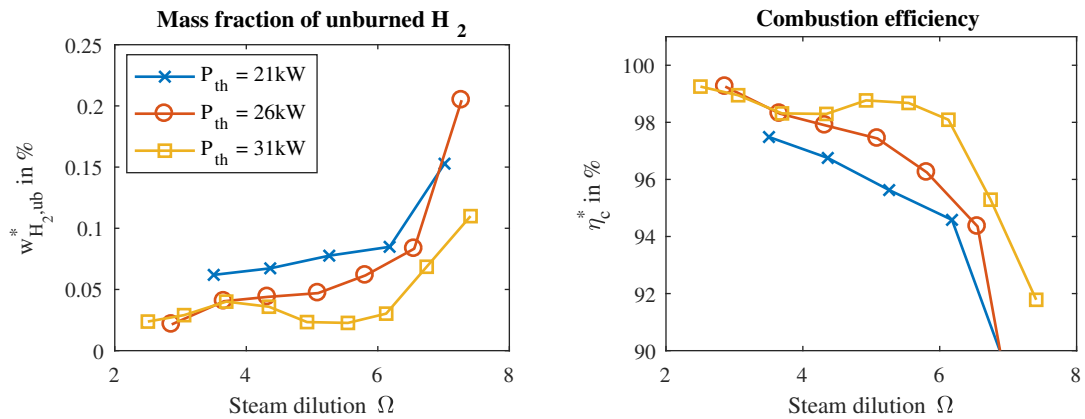


Figure 5.15: Left: mass fraction of unburned H₂ calculated by Eq. (5.15). Right: combustion efficiency calculated by Eq. (5.16). Both for burner Configuration III.

Thus, the impairing effects are already dominant when reaching the swirl-stabilized mode. At $\Omega \approx 6.5$, the combustion efficiency drops significantly for all power levels. At this stage, the flame is very receptive to fluctuations of the steam mass flow, which leads to a fluctuation of the combustion efficiency. Moreover, the flame is close to cold blow-off, a condition not suited for industrial applications. However, the most suitable application is located in the range of $\Omega = 3.5$ -5 (see Fig. 4.20), where the 25 and 30 kW flames show a good combustion efficiency.

Please note that the quantification of the combustion efficiency in this section relies to some extent on the accurate measurement of the equivalence ratio by the Coriolis mass meters. The beginning of Sec. 5.5 comments on this issue.

5.4.2 Variation of Burner Configuration

As found in the previous section, Configuration III shows a local maximum of the combustion efficiency because of the positive effect of the swirl-stabilization. As shown in the isothermal investigations (Chapter 3), the swirl intensity can be increased by changing the burner configuration. By using Configuration XI, the swirl number is approximately doubled (Fig. 3.15). As a trade-off, Configuration VII was chosen for investigation. It also had an increased swirl intensity, but only exhibited a moderate degradation of the pressure loss. The increased swirl intensity also reduced the steam dilution ratio at which the flame shape transition took place (see Fig. 4.21). For instance, using Configuration XI, the flame was already in the transitional mode at the lowest investigated steam dilution ratio. It did not reach the jet mode.

Again, the O_2 concentrations in the exhaust are measured by the TDLAS O_2 sensor. At the lowest steam dilution, all three configurations show the same O_2 residuals in Fig. 5.16. While for Configuration III they increase in the transitional phase, they stay constant for Configuration VII. For Configuration XI, they decrease immediately as this is the first flame establishing the swirl-stabilized mode. At $\Omega = 4.5$ the TDLAS sensor measured a residual concentration of $w_{O_2,ex} = 0\%_m$ within its uncertainty. As soon as the flame is in the swirl-stabilized mode, all the configurations show comparable concentrations until the efficiency drop sets in at $\Omega = 6.5$. The dropping point is unaffected by the burner configuration because it is caused by the thermal flame stability. As such, it is independent of the swirl intensity as long as all flames are in the swirl-stabilized mode.

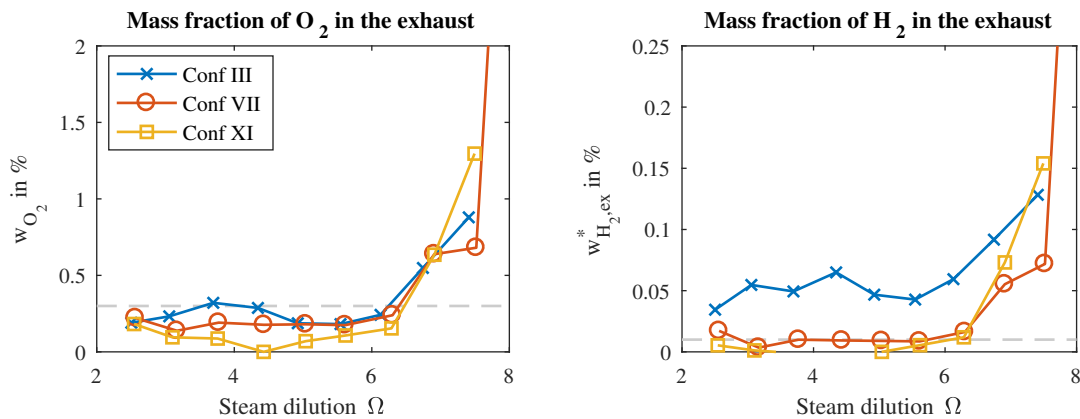


Figure 5.16: Residual gas mass fractions in the exhaust at 30 kW thermal power. Left: O_2 measured by the TDLAS sensor. Right: H_2 calculated by Eq. (5.17). The dashed lines mark the target concentrations in the DLR project [20].

In contrast to the observations in Sec. 5.4, the H_2 concentrations, shown on the right side of the figure, do not match the O_2 ones. The flame of Configuration III is measured as slightly rich while the other two are slightly lean (see Fig. C.1). Consequently, the H_2 concentration of Configuration III in the exhaust is much higher. On the other hand, for Configuration XI, the values of $w_{H_2,ex}^*$ are reduced by the nonstoichiometric part $w_{H_2,phi}^*$ according to Eq. (5.17). This can lead to negative and, therefore, unphysical H_2 concentrations in the exhaust. The same holds for the mass fraction of unburned stoichiometric H_2 and, eventually, yields to a combustion efficiency larger than 100 %, as shown in Fig. 5.17 at $\Omega = 4.5$ for Configuration XI. This result seems unrealistic but stems from the uncertainty of the Coriolis mass meters, which will be discussed in Sec. 5.5.

Configuration VII and XI reveal a very high combustion efficiency over a broad spectrum from $\Omega = 2.5 - 6.5$. The efficiency of Configuration XI is slightly higher, in the technically most relevant range of $\Omega = 3.5 - 5$, due to the increased swirl intensity. Configuration III shows a continually lower combustion efficiency except for the point of lowest steam dilution (jet flame). At this point, the adiabatic flame temperature is very high, and since none of the flames are purely swirl-stabilized at this condition, the swirl intensity does not seem to have such a strong influence. For increased steam dilution, Configuration III is the only one with a local minimum during the flame transition, which is smoother and more stable with higher swirl intensity. If the swirl intensity is too low (e.g., Configuration I with saturated steam), the flame can even become unstable and lead to flame lift-off during the transition.

Please note that the quantification of the combustion efficiency in this section relies on the measurement of the equivalence ratio by the Coriolis mass meters.

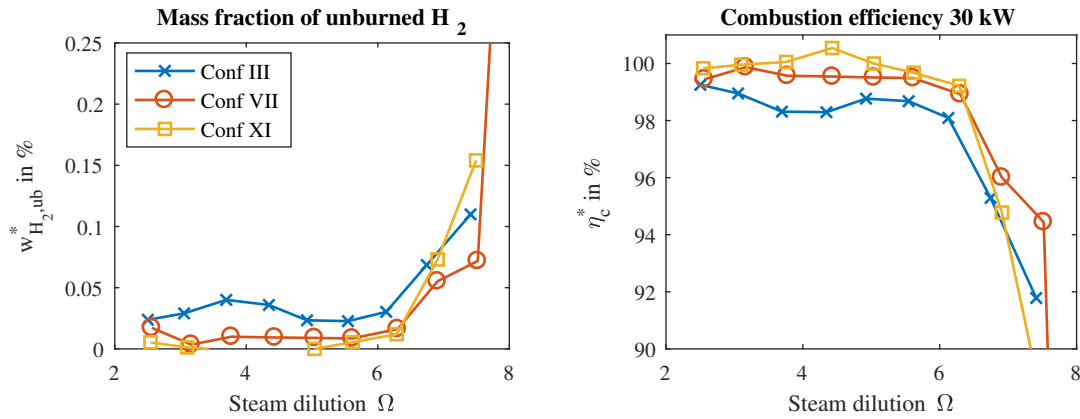


Figure 5.17: 30 kW thermal power: Left: Mass fraction of unburned H_2 calculated by Eq. (5.15). Right: Combustion efficiency calculated by Eq. (5.16).

5.5 TDLAS- and LS-based Combustion Efficiency Measurements

The results given in the previous section are based on the measurement of the equivalence ratio via Coriolis mass meters to ensure a stoichiometric reactant ratio. However, small deviations in the H_2 mass flow can have a significant impact, which will be discussed in a separate subsection here. The alternative to the previous approach is the determination of the equivalence ratio using the LS, as described in Sec. 5.3.3. The combustion efficiencies, corresponding to this approach, will be presented afterwards.

Uncertainty of the Equivalence Ratio

Figure 5.18 shows the equivalence ratios measured with both methods (LS: solid line, Coriolis: dashed line) for flames at different power levels. On the left side, the reactant mass flows were set to the stoichiometric ratio s according to the Coriolis mass meter readings. In this case, the LS indicated that the flame was rich. On the right side, the H_2 mass flow was fixed, while the LS-coupled PID controller set the O_2 mass flow to match the LS threshold voltage. In this case, the equivalence ratio calculated from the mass meters was below one. According to the test in Fig. 5.9 (right), the catalytic supported LS provides the stoichiometric jump at the correct H_2/O_2 ratio. Hence, the H_2 mass flow measured by the Coriolis mass meter was too small in the experiments, which indicates that the flames in the previous section were not stoichiometric but slightly rich. As a consequence, the observed O_2 content in the exhaust was very low because it was more H_2 available than needed to convert all the O_2 . As a result, the H_2 concentration in the exhaust was probably higher than reported and overestimated the combustion efficiency.

The Coriolis mass meters used for the monitoring of the reactant mass flows had a nominal uncertainty of 0.5 % of the measurement value. The size was chosen according to the guidance of the manufacturer as DN4 for both gases. The uncertainties are given in Fig. 5.19 as a function of the thermal power. As intended, it is 0.5 % for O_2 . However for H_2 , the uncertainty is much higher because of the low H_2 mass flows, which were eight times smaller than for O_2 . The figure also displays the uncertainty of CH_4 , which is much smaller than for H_2 and underlines problematic behavior of the H_2 measurement. For CH_4 , a smaller mass meter (DN2) could be used, which is not possible for H_2 because its low density causes very high velocities in the measurement section. In order to confirm a systematic error, two Coriolis mass meters were mounted sequentially and found to differ from each other in the order of the deviation given in Fig. 5.19. This uncertainty of the reactant mass flows translates into an uncertainty of the equivalence ratio of approximately 0.05 in the investigated power range of 20-30 kW. It explains the difference of the equivalence ratios between the LS and the Coriolis measurements in Fig. 5.18.

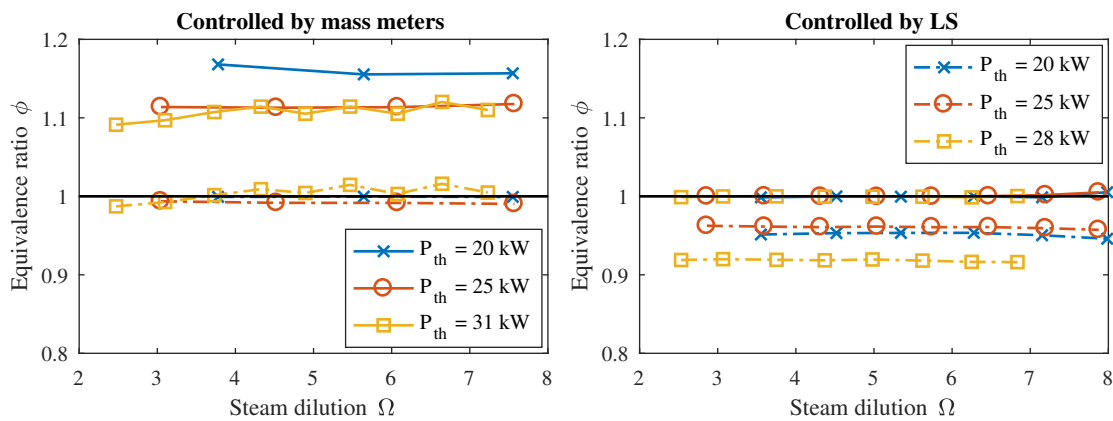


Figure 5.18: Equivalence ratio during combustion tests with Configuration VII calculated from the LS (solid line) and the mass meters (dashed line). Left: Mass flows set according to mass meters. Right: Mass flows set according to LS.

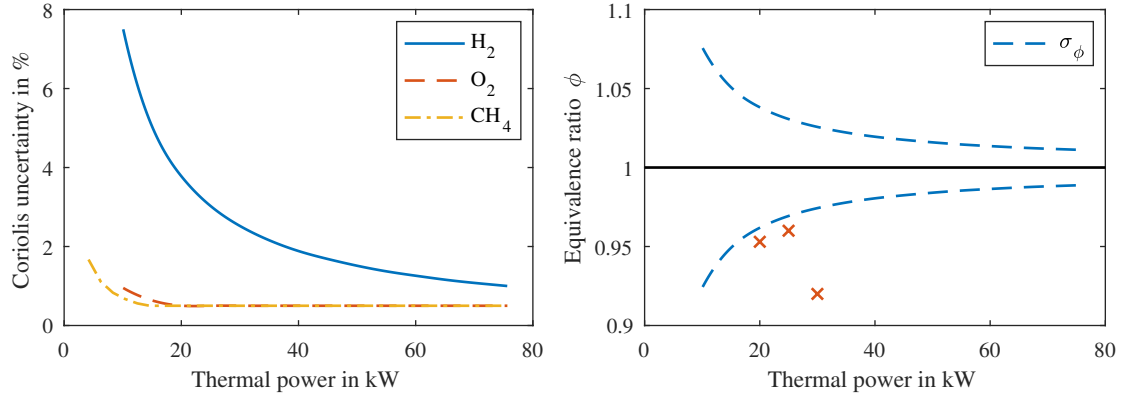


Figure 5.19: Left: Measurement uncertainty of the Coriolis mass meter for H_2 (DN4), O_2 (DN4) and CH_4 (DN2) [57]. Right: Relative uncertainty of the equivalence ratio for $\phi = 1$ calculated by Eq. (D.1). The crosses show the LS measurements from Fig. 5.18.

Combustion Efficiency Measurements with the LS-based Equivalence Ratio

The results in the current section refer to flames which are controlled by the LS. Again, the O_2 mass fractions in the exhaust were measured by the TDLAS O_2 sensor. They are shown in Fig. 5.20 for a thermal power level of 30 kW. Compared to Fig. 5.16, the residual O_2 concentrations are much higher because the PID controller increased the O_2 mass flow to ensure $\phi_{LS} = 1$. However, the general trends described before remain the same. This time, the deviation from stoichiometry is calculated from the signal of the LS instead of the mass meters. Since the O_2 mass flow was controlled accordingly, the contribution of $w_{O_2,\phi}$ and $w_{H_2,\phi}$ to the residual gas concentration is negligible, as shown in Fig. 5.21.

Hence, the residual gas components in the exhaust result dominantly from combustion inefficiencies, and the H_2 concentrations equal the O_2 ones in Fig. 5.20 expect for the stoichiometry factor s . For all configurations, the residual gas concentrations in the exhaust are low for a steam dilution ratio of $\Omega \approx 2.5$. Increasing Ω brings the flame from the jet into the transitional mode and decreases the adiabatic flame temperature. The residual gas fractions increase to a local maximum. Afterwards, they are decreasing again to a local minimum before the flame weakening towards blow-off sets in. This effect is the strongest at Configuration I, which has the lowest swirl number (see Fig. 3.15). The local maximum shifts towards lower steam dilution ratios for increasing swirl numbers, and at Configuration XI, the maximum is not prominent anymore. Here, the O_2 concentration in the exhaust is below 0.52 %_m over the whole range of measured steam dilution ratios with a minimum in 0.39 %_m. The minimum H_2 concentration is below 0.05 %_m.

The mass fraction of unburned H_2 (Fig. C.2) is nearly the same as $w_{H_2,ex}$ due to the extremely low deviations from the stoichiometric ratio. Thus, the remarks of the previous paragraph are also valid for the combustion efficiency, which is calculated from $w_{H_2,ub}$ by Eq. (5.10) and presented in Fig. 5.22 (right). For all configurations, the combustion efficiency reaches 98-98.5 % at the lowest steam dilution ratios, respectively, the highest adiabatic flame temperature. The Configurations I, III, and VII show a local maximum of the combustion efficiency after reaching the swirl-stabilized mode around $\Omega = 5$. Thereby, the efficiency is higher as the swirl number of the burner increases. Configuration XI reveals no local extrema, but the efficiency is nearly constant around 97 % in the range of $\Omega = 3.5$ -5.

In contrast to the observations without the LS in Fig. 5.17, the efficiency does not drop significantly at the highest steam dilution ratio. The reason is that the equivalence ratio

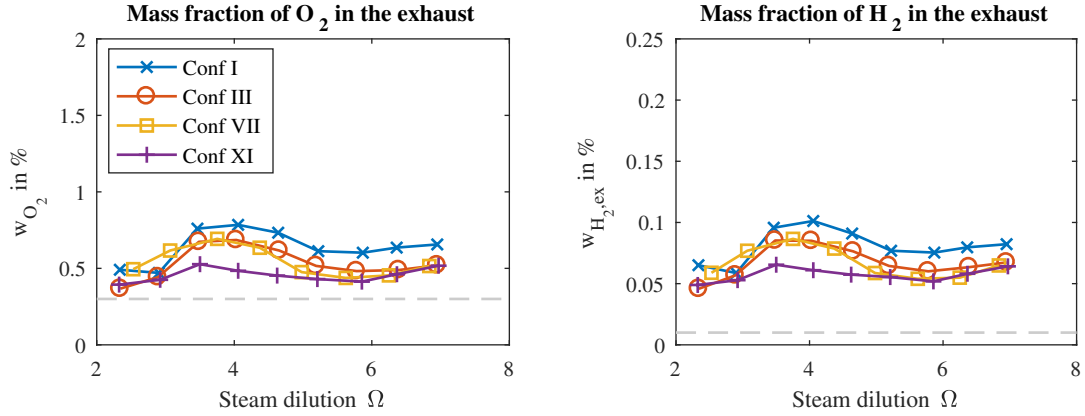


Figure 5.20: Residual gas mass fractions in the exhaust at 30 kW thermal power. Left: O₂ measured by the TDLAS sensor. Right: H₂ calculated by Eq. (5.11). The dashed lines mark the target concentrations in the DLR project [20].

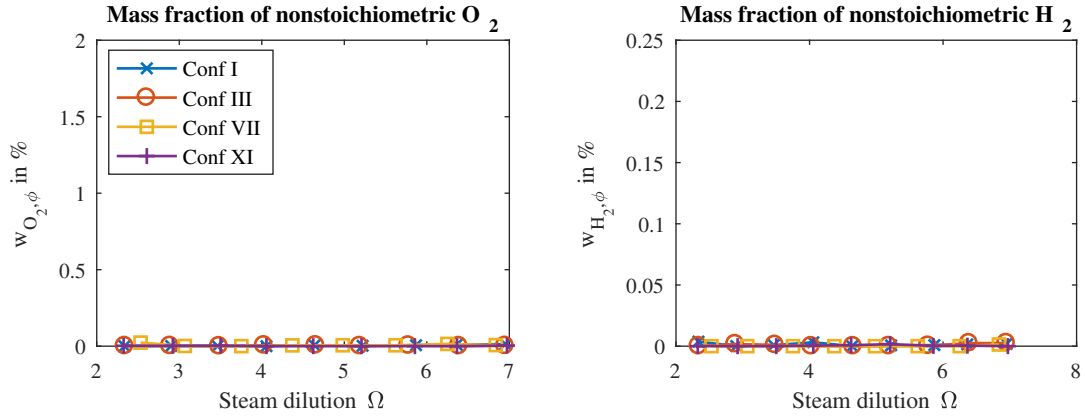


Figure 5.21: Deviation from the stoichiometric ratio in mass fractions for burner at 30 kW thermal power. Left: O₂ measured by the LS and Eq. (5.4). Right: H₂ measured by the LS and Eq. (5.5).

was set more accurately to stoichiometric conditions. In the previous case, the flame was rich, which means that not enough oxygen was available to convert all the fuel. Hence the effective thermal power was lower. Applying the correct H₂/O₂ ratio allows the flame to withstand the high steam dilution even better, and shifting the efficiency drop to higher Ω. It now lies outside of the measured range. However, the efficiency drop still exists, as Fig. 5.22 (left) proves for a lower thermal power of 20 kW. These low power flames also show a much stronger dependency on the swirl number.

For each configuration, the combustion efficiency depends on the thermal power density and is higher for 30 kW. It is expected that the combustion efficiency gets closer to 100 % for higher power densities. Section 4.4 proves that stable operation of the combustor is also possible with 70 kW in the same combustion chamber. This means that the power density would be more than doubled, which could increase the combustion efficiency. Unfortunately, increasing the power level in CTR B was not possible in the experiments due to limited reactant flow rates. Please note that the cited experiments of the DLR with the H₂/O₂-SG, taken as benchmark, were conducted at a thousand times higher thermal power level. By contrast, the cross section of the combustion chamber was only 56 % larger yielding a much higher power density.

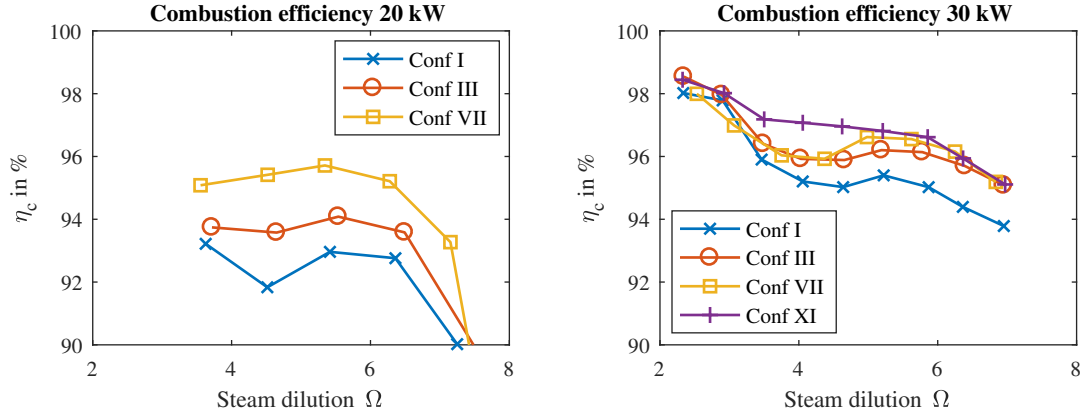


Figure 5.22: Combustion efficiency calculated by Eq. (5.10) for 20 kW (left) and 30 kW thermal power.

Influence of the Combustion Chamber Material

The experiments in this thesis required two kinds of combustion chambers: a fused silica one for the flame shape observations and a steel one for the heat transfer measurements. At last, the influence of these different combustion chamber materials was measured exemplarily at Configuration III. The steel combustion chamber was cooled by a co-axial 25 kg/h air flow with an inlet temperature of approximately 293 K (see Sec. 4.5). The silica combustion chamber, on the other hand, was not actively cooled but surrounded by ambient air.

Figure 5.23 shows the combustion efficiency for both cases at 25 and 30 kW. At steam dilution ratios less than three, there is no difference. A jet flame establishes close to the central axis of the combustion chamber. In this case, the flame is shielded from the chamber walls by a thick layer of steam. Thus, the heat loss has less effect on the combustion efficiency. As the steam dilution ratio increases, the flame angle spreads (see Fig. 4.19), and the reaction zone moves closer to the wall. In this case, the combustion efficiency of the steel combustion chamber is continuously higher, approximately 2 % for 25 kW and 1 % for 30 kW. An intuitive conclusion would be that the combustion efficiency of the fused silica combustion chamber is lower due to an increased heat loss, which causes a lower temperature in the combustion chamber. The effect of a lower temperature on the combustion efficiency can be seen in Fig. 5.23 by comparing the 25 (left) and the 30 kW case (right). Figure 5.24 points out a significant lower exhaust temperature in the 25 kW case due to the lower adiabatic flame temperature. Simultaneously, the combustion efficiency declines.

However, the comparison of the different combustion chambers shows an opposite effect. The exhaust of the silica chamber is hotter (Fig. 5.24) and thus has a lower total heat loss. Nevertheless, the combustion efficiency is lower. This behavior is observed at both thermal power levels. The higher temperature is justified by the thermal conductivity of fused silica which is about one order of magnitude lower than the one of stainless steel (4571). However, in contrast to steel, the fused silica permits radiative heat loss, which originates directly in the reaction zone and might be responsible for increased quenching. The share of radiation on the heat flux is much higher for a stoichiometric H_2 oxyfuel flame than for air flames due to the high steam concentration [8, 16, 19, 22]. It could explain the slightly lower combustion efficiency. The heat transfer in the steel combustion chamber was analyzed in Sec. 4.5.

Please note that Figure 5.23 shows the uncertainty of the efficiency measurements as errorbars, exemplarily for the plots in the current chapter. The corresponding uncertainty quantification is discussed in the Appendix D.

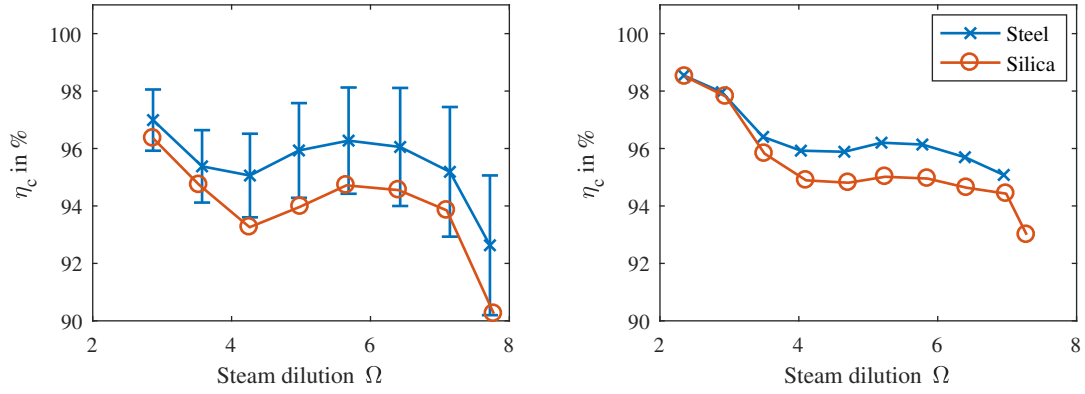


Figure 5.23: Combustion efficiency calculated by Eq. (5.10) with a air-cooled steel and a fused silica combustion chamber at burner Configuration III with 25 kW (left) and 30 kW (right).

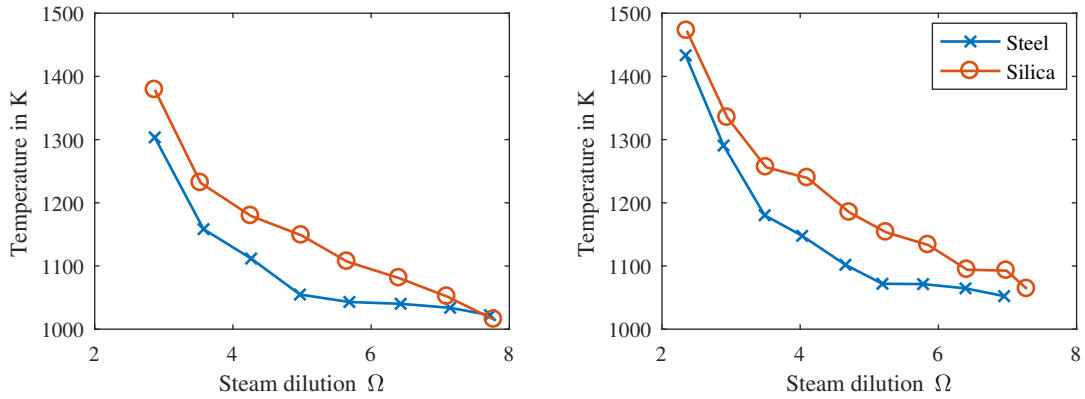


Figure 5.24: Exhaust temperature with a air-cooled steel and a fused silica combustion chamber at burner Configuration III with 25 kW (left) and 30 kW (right).

5.6 Summary

The combustion efficiency of the H_2/O_2 burner was evaluated by measuring the O_2 concentration in the exhaust based on a TDLAS sensor. The investigations were conducted with two approaches to control the equivalence ratio. First, the reactant flow rates were set to match the stoichiometric ratio s according to the mass meter readings. Second, the LS controlled the equivalence ratio to $\phi_{\text{LS}} = 1$ automatically.

In the first case, the results depended strongly on the accuracy of the mass meters. They caused a considerable uncertainty of the equivalence ratio. As a result, the flames seemed to be more rich than stoichiometric because the Coriolis underestimated the H_2 mass flow. This error resulted in an overestimation of the combustion efficiency. However, the qualitative findings and trends were similar to those of the second approach.

The LS-based approach took profit of the high sensitivity of the LS at stoichiometric conditions. Therefore, it was much more suited to control the equivalence ratio precisely. With $\phi = 1$, the residual gas concentrations in the exhaust of the flame stemmed mostly from combustion inefficiencies and not from deviations of the stoichiometric ratio. The current study found that a heated catalyst was required for sufficient pre-oxidation. However, this

might depend on the specific system purchased. In any case, the behavior of the LS has to be tested with a mixture of H_2 and O_2 in steam.

The combustion efficiency depended on the power density and increased with raised thermal power. It is expected that further enhancement of the thermal power density would further improve the combustion efficiency. Generally, the efficiency was high up to a certain steam dilution ratio. If the flame was further diluted, the efficiency dropped significantly. The drop occurred close to the cold blow-off limit, but its specific dilution ratio was independent from the burner configuration, i.e., the swirl intensity. However, it depended on the equivalence ratio as stoichiometric flames could withstand the dilution better than nonstoichiometric ones.

All configurations showed the highest combustion efficiency for the lowest steam dilution ratio. Here, the efficiency reached $\eta_c = 98\text{--}98.5\%$. In the transition between the jet and the swirl-stabilized mode, there was a slight local minimum of the efficiency (except Configuration XI). The swirl-stabilized range showed an elevated efficiency again. Here, the efficiency depended on the swirl-intensity. Hence, Configuration XI (with the highest swirl number) yielded the best combustion efficiency of $\eta_c = 97\%$ in the swirl-stabilized mode. Due to the burner design, it was not possible to investigate a jet flame in this high-dilution range for comparison. This task is left open for future research.

A comparison of the combustion chamber material showed that the air-cooled steel tube featured a higher combustion efficiency than the fused silica one, even though its exhaust temperature was lower. It is supposed that the radiative heat transfer was responsible for this effect.

CHAPTER 6

Conclusions and Outlook

The combustion of H_2 and O_2 is considered a promising way of environmentally friendly power generation. It provides a zero-emission back-conversion of the energy carrier H_2 into heat and electric power. The thesis introduced two applications for an H_2/O_2 combustor. In the short-term perspective, PCR provision was shown to be economically viable and offers the potential to be combined with wind or solar farms. In the long term, a zero-emission power plant using the H_2/O_2 Graz cycle is an excellent option to utilize stored H_2 . The studied energy storage system presented an extraordinary thermal and exergetic efficiency.

Previous research projects on H_2 oxyfuel combustion were based on rocket technology with downstream water vaporization. In contrast, the current study aimed to adapt the design of a low-emission gas turbine combustor. Therefore, the concept of swirl-stabilized combustion with upstream steam dilution was applied. Since there was almost no literature on the utilization of this concept for H_2 oxyfuel combustion, the thesis presented the development step by step: from a fuel-flexible air burner to an H_2/O_2 combustor. The thesis can, therefore, act as a blueprint for future combustor developments.

Once the final design for the lab-scale burner was chosen, a full characterization was performed in the water tunnel and the combustion test rig. The burner featured non-swirling reactant nozzles and two coaxial ducts for swirled steam injection. Hence, the swirl number of the flow in the combustion chamber depended on the steam dilution ratio. This specific design revealed an interesting behavior: two distinct flow and flame types were observed depending on the operating point.

At low steam dilution, the steam and the reactant flows merged to an axial jet. The reactants were mixed in the SLs of this jet by turbulent diffusion. The mixture was convected downstream and immediately burned in a longish, cylindrical jet flame.

At high steam dilution, the swirl intensity of the flow was sufficient to enable vortex breakdown and form an inner recirculation zone in the center of the combustion chamber. The stagnation line of the inner recirculation zone stopped the propagation of the H_2 . Instead of being convected downstream along the central axis, the H_2 was transported outwards to the shear layers of the recirculation zone. Here, it mixed with the O_2 and combusted with the characteristics of a swirl-stabilized flame. It was significantly shorter and spread more in the radial direction. In general, the swirl-stabilized mode presented a faster and more homogeneous mixing of the reactants, which is favorable for the combustion efficiency.

The transition between both modes ran smoothly with a variation of the steam dilution ratio Ω . The critical Ω depended on the thermal power, although the swirl number did not, as shown in the CFD analysis. The explanation for this effect is that the flame's heat release generated vorticity and strengthened the backflow in the inner recirculation zone. Hence, the transition appeared earlier at higher thermal power. As a result, the water tunnel findings

cannot be directly translated to the flame in terms of the exact Ω . Nevertheless, a good consistency between the flow, the mixing, and the flame behavior was found throughout the study. The water tunnel measurements of the flow and mixing field proved to be accurate tools for the analysis and prediction of the H_2 oxyfuel flame, as previously reported by other authors for natural gas-air flames.

Besides defining the flow and flame shape, the steam dilution facilitates straightforward control of the adiabatic flame temperature, which is necessary for the process integration of such a combustion system. However the range of steam dilution is limited. In the current study, the burner's operational limits were given by the extreme heat load on the combustor at low steam dilution and by the cold blow-off limit at high steam dilution. On the lower limit of the operating range, flames with $\Omega = 1.1$ and $T_{\text{ad}} = 2483 \text{ K}$ were operated without any damage to the burner or the combustion chamber, which shows the robustness and potential reliability of the applied combustion concept. The upper limit was found to be at $\Omega \approx 7 - 9$ or an adiabatic flame temperature of 1000-1100 K, respectively. The critical Ω for blow-off was found to decline with increasing thermal power due to the increased bulk flow velocity. This reduction of the operating range has to be respected when increasing the burner's power density.

If the burner is integrated into a power cycle, the upper limit of steam dilution is further limited by the combustion efficiency. When the reactants were diluted too strong, the combustion efficiency dropped, and unburned residuals of H_2 and O_2 remained in the exhaust, which could damage downstream components of the power cycle. In order to avoid these, the steam dilution ratio should be kept below 6.5.

The hazard of these unburned gas residuals requires online monitoring of the combustion efficiency. Therefore, a time-resolved combustion efficiency measurement system was deployed in this thesis for the first time for H_2/O_2 combustion. The method, based on absorption spectroscopy of O_2 in the exhaust, demonstrated to work well if an LS is operated simultaneously to control the stoichiometry. It was found that controlling the equivalence ratio with Coriolis mass meters was not sufficient in terms of the high accuracy needed to avoid gas residuals in the pure product steam of an H_2/O_2 flame.

The maximum combustion efficiency, reached in the study, was about 98-98.5 % at the lowest measured steam dilution ratio, i.e., at the highest flame temperature. During the transition from the jet to the swirl-stabilized flame, the combustion efficiency decreased to approximately 96 %. In the swirl-stabilized mode, it showed a local maximum again. This region is considered an optimal operating point.

The O_2 concentration in the exhaust, resulting from the small combustion inefficiencies, was very close to the DLR benchmark value. The H_2 concentration, on the other hand, was much higher than this threshold. A solution could be to balance the residual concentrations by burning slightly lean, but the experiments indicated that a non-stoichiometric mixture impairs the combustion efficiency, especially in the highly diluted range. Another way to reduce the residual concentrations is to increase the power density of the combustion chamber. The results presented a higher combustion efficiency for increased thermal power. However, the air, used as cooling flow in the thesis, would not be sufficient for the active cooling of the combustion chamber at increased power density, as shown by the heat transfer measurements. Hence, the author proposes regenerative cooling by the diluent steam for future applications.

Another influence on the combustion efficiency was found to be the combustion chamber material. The stainless steel tube turned out to be favorable over the fused silica one. This issue is important for translating results from the lab, where silica glass is often used to gain optical access, to the technical applications, which usually utilize steel chambers.

Next to these general remarks on the steam diluted H_2/O_2 combustion, a parametric

design study has been performed within the thesis. The results of this study are summarized in Fig. 6.1. Further blockage of the swirl generators at higher burner configurations primarily increased the swirl number in the combustion chamber, which resulted in a widening of the flow field and an upstream shift of the IRZ. As expected, the increased swirl intensity improved the mixing quality. Additionally, it was found to enhance the combustion efficiency in the swirl-stabilized mode. Solely Configuration I with non-preheated steam showed an unstable flame type transition. The configurations with increased swirl intensity were found to be stable in the entire operating range. Moreover, Configuration XI presented the broadest operation in the swirl stabilized mode. For these reasons, the swirl intensity should be as high as possible. However, these benefits come at the cost of increased pressure loss. It is shown inverted in Fig. 6.1, such that Configuration I has the lowest and XI by far the highest pressure loss. Configuration VII seems to be a good trade-off between all considerations.

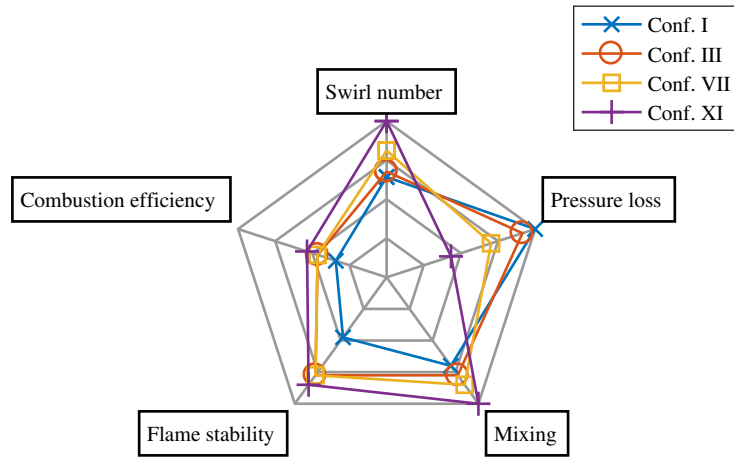


Figure 6.1: Evaluation of the burner configurations: effect on the swirl number (normalized by Conf. IX), the pressure loss ($(\Delta p/q)_I / (\Delta p/q)$), the mixing and the flame stability (both evaluated qualitatively), and the Combustion efficiency (range 95-100 %).

The two-staged steam injection via the inner and outer steam ducts has been proven to be a suitable design in the experiments. No burner or combustion chamber wall damage has been detected, nor any accumulations of O_2 in the outer recirculation zone. Moreover, it gives a tool to tailor the flow field in the designing phase of the combustion system, as it has been demonstrated in the parameter study.

In summary, the thesis proved the successful operation of gas turbine combustor technology for H_2 oxyfuel combustion. The concept of steam-diluted H_2/O_2 combustion should be further developed and brought to the application level, especially as H_2 is seen more and more as the future's primary energy carrier. The last part of the thesis, therefore, presents a brief outlook with recommendations for future investigations.

For H_2/O_2 combustor applications, it is recommended to change the burner design, so it generates either a jet or a swirl-stabilized flame in the entire Ω range. The first can be done by removing the swirl generator and injecting the steam as a non-swirling axial flow. However, this design would result in an elongated flame due to impaired mixing. The second can be reached by adding a swirl generator to the H_2 nozzle and tilting the O_2 jets, as it was done in the preliminary version of the burner. This measure would have multiple benefits. First, it would increase the swirl intensity and, hence, the combustion efficiency, as presented in the thesis. Second, by tuning the tangential momentum of all three flows (H_2 , O_2 , and steam), the flow field would no longer depend on the steam dilution ratio. Instead,

a constant flame shape and position can be expected throughout the entire operating range. Finally, with swirled reactant flows, the same amount of swirl intensity can be reached with less blockage of the swirl generators, which would reduce the pressure loss.

The TDLAS-based measurement approach, presented in this thesis, showed to be suitable for online monitoring of the combustion efficiency. However, the uncertainties could be reduced by a more sophisticated infrastructure. The following recommendations are made from the lessons learned during the work for this thesis.

As the operating conditions in a specific application will be narrow, the TDLAS sensor should be calibrated at exactly these conditions (temperature and pressure). This way, the uncertainty of corrections will be avoided. At larger combustor dimensions, the TDLAS measurement should be conducted in a cross-section of the exhaust tube rather than in an additional measurement tube. Thus, the measurement will be locally, and the conditions in the optical pathway will be more constant, increasing the accuracy. However, even at the 70 MW demonstrator of the DLR, the diameter was not significantly larger. Mirrors could be used instead to reflect the laser beam multiple times through the cross-section, thus increasing the optical path length. Such a design makes the system more complex and harder to align, but minimizes the lower detection limit and improves the accuracy. Finally, if the system is operated at high pressures, the sensor heads must be protected by windows between the purge and the exhaust gas. This measure would have the additional benefit that the signal is not influenced anymore by changes in the mass flow, as it has been reported in the thesis. The implementation of these improvements will result in a robust system for accurate and time-resolved monitoring of the combustion efficiency.

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APPENDIX A

Burner Inlet Temperatures

The reactant temperature at the inlets is essential because it determines the density and the viscosity, which enable the calculation of the Reynolds number and the bulk velocity of the reactant jets. These values were used in the study as characteristic quantities. However, the temperatures of the reactant gases were not measured in the experiments due to the small and complex geometry of the combustor. The fuel was supplied via the burner flange at three ports (2 O₂, 1 H₂). From there, the reactant lines ran through the plenum and were attached to the burner's upstream end. In the plenum and inside the burner, the walls of the lines were heated up by the super-heated steam. In order to estimate the temperature, numerical simulations of the reactant supply were conducted in this chapter.

A.1 Numerical setup

The geometry of the burner's fuel supply system is shown in Figure A.1. In the simulations, the reactant flow was modeled from the flange outlet to the inlet of the combustion chamber. The

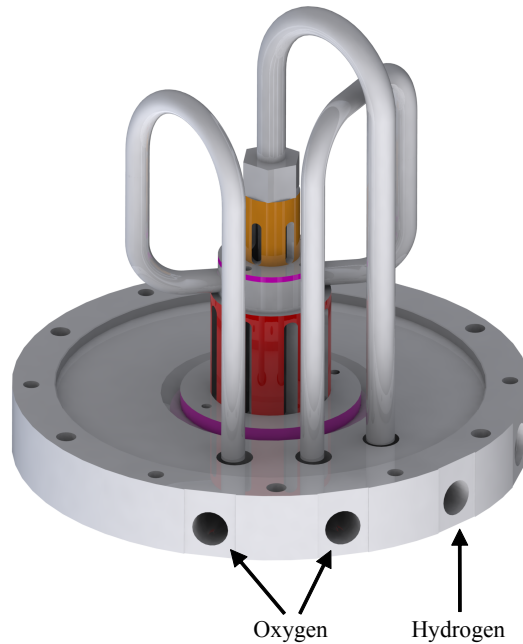


Figure A.1: Combustor with fuel lines.

simulations were conducted with *CFX* using steady-state RANS and a $k\Omega$ -SST turbulence model. The heat transfer from the wall to the fluid was modeled by the heat flux through the wall

$$q_w = h_c (T_o - T_w), \quad (\text{A.1})$$

using an external heat transfer coefficient h_c , the outside temperature T_o (i.e., the steam temperature) and the temperature of the inner wall surface T_w [10]. The external heat transfer coefficient was calculated individually for each part according to its geometry and material:

$$h_c = \frac{k}{r_i \cdot \ln\left(\frac{r_o}{r_i}\right)}. \quad (\text{A.2})$$

Table A.1 shows the inner radius r_i , the outer radius r_o , the thermal conductivity of the material (stainless steel), and the calculated h_c . The heat transfer coefficient was not calculated at the O_2 burner nozzles because they have only a length of 4 mm. Therefore their influence was considered negligible compared to the other parts. Instead, a fixed temperature was chosen as the BC here. The simulations were conducted for two reference cases with a thermal power level of $P_{\text{th}} = 50 \text{ kW}$ and $T_{\text{steam}} = 373, 573, \text{ and } 673 \text{ K}$, respectively. The inlet conditions are given in Tab. A.2. The mesh contained approximately 1.47 million elements for the O_2 line and 0.53 million for the H_2 line. The walls were refined by prism layers to reach $y^+ < 40$.

Table A.1: External heat transfer coefficient

	r_i [mm]	r_o [mm]	k [W/mK]	h_c [W/m ² K]
H_2 plenum pipe	4.5	6	13	10000
H_2 burner pipe	4.5	5.5	13	14400
O_2 plenum pipe	4	5	13	14600
O_2 burner annulus	9 / 11.35	9.65 / 12	13	20600
O_2 burner nozzles	0.75	-	13	-

Table A.2: Inlet conditions.

	H_2	O_2	
Mass flow	1.49	11.9	kg/h
Velocity	70.7	24	m/s
Temperature	293	293	K
Turbulence intensity	5	5	%

A.2 Results

All the four simulations, conducted in this numeric pre-test, converged sufficiently. Figure A.2 shows the streamlines of both reactants in the case with saturated (left) and super-heated (right) steam. The streamlines, as well as the inlet and outlet surfaces, are colored by the gas temperature. Table A.3 presents the area-averaged values of velocity, temperature, and density. In both cases, there was only a small temperature difference between the H_2 and O_2 injection. The Reynolds numbers were calculated using the simulated velocity and temperature, as well as the viscosity from the NIST database [116]. They were larger for

the H_2 injection, but generally in the same order of magnitude. Even though the velocity increased in the super-heated case, the Reynolds numbers were lower due to the increased viscosity.

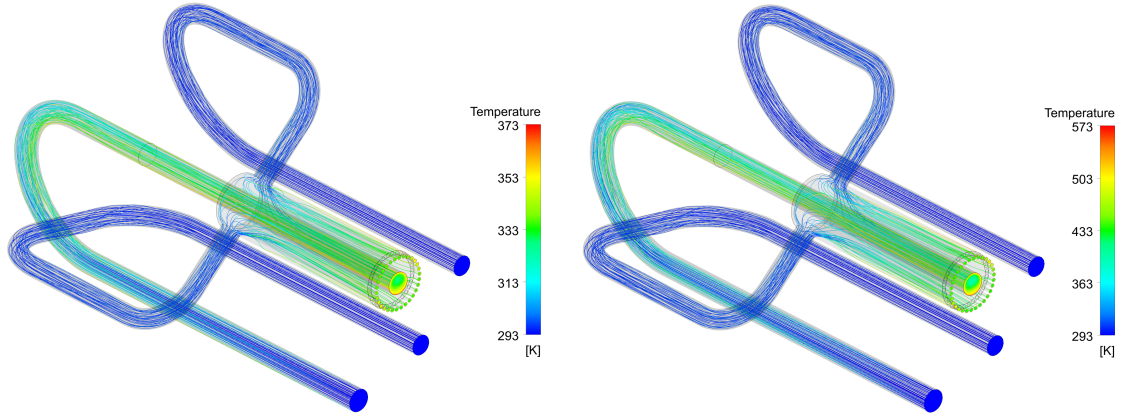


Figure A.2: Fluid temperature in the fuel lines for a plenum steam temperature of $T_{\text{steam}} = 373 \text{ K}$ (l.) and $T_{\text{steam}} = 573 \text{ K}$ (r.).

Table A.3: Outlet area average.

	H_2	O_2	
$T_{\text{steam}} = 373 \text{ K}$			
Velocity	91.0	55.6	m/s
Temperature	352.5	347.0	K
Density	0.0697	1.125	kg/m^3
Reynolds number	5640	4000	
$T_{\text{steam}} = 573 \text{ K}$			
Velocity	118.8	74.1	m/s
Temperature	458.0	466.1	K
Density	0.0540	0.841	kg/m^3
Reynolds number	4740	3180	
$T_{\text{steam}} = 673 \text{ K}$			
Velocity	131.5	83.3	m/s
Temperature	512.6	526.5	K
Density	0.0486	0.746	kg/m^3
Reynolds number	4340	2890	

APPENDIX B

TDLAS Purging Simulation

In order to assure reliable measurements with the TDLAS system, the purging flows had to be adequately designed to obtain a fixed optical path length (L_{op}). A CFD analysis was carried out to get insights into the flow patterns since the measurement section provided no optical access and was not accessible by flow sensors. The numerical investigation focused on the union cross, where the exhaust stream faced the N_2 purge (see Figure 5.4).

B.1 Numerical Setup

The simulations were conducted in the steady-state RANS framework using the commercial flow solver *CFX* with the well known $k\Omega$ -SST turbulence model and an automatic wall function. The fluid contained three species with different temperatures and enabled diffusion. However, the heat transfer between them was neglected: each fluid was treated as isothermal. The computational domain was reduced to the necessary parts compared with the setup shown in Figure 5.4. Since the TDLAS section is nearly symmetrical, just one side was simulated. The inlet of the steam was set upstream of the main outlet tee in the measurement section (see Figure B.1). A uniform velocity profile was assumed here, which might not be the case in real operation. However, it was not expected to change the flow in the union cross significantly. At the outlet of the TDLAS section, a bent pipe was added in accordance with the test rig. The pipe was narrowed downstream of the tee in the straight direction, where the steam entered the union cross. The cross had three inlets and one outlet. The upper inflow consisted of preheated N_2 , while the right one was N_2 at room temperature. A bent pipe was attached to the outlet port just as for the main steam outlet. The BCs are given in Tab. B.1.

Table B.1: Boundary conditions of the purging flow simulations.

	Steam inlet	Hot purge	Cold purge	
Medium	Steam	N_2	N_2	
Mass flow	10 – 25	0.23 – 0.34	0.23 – 0.34	kg/h
Temperature	973	473	293	K
Density	0.23	0.72	1.17	kg/m ³
Turbulence	5	5	5	%

The steam mass flow was varied to represent different steam dilution ratios in the experiment. Additionally, the purge mass flows were varied to find the purging rate that stopped the hot exhaust from draining into the cold section, but at the same time did not dilute the exhaust in the measurement section.

The mesh consisted mainly of tetrahedra cell with three prism layers on the wall. The total number of elements summed up to approximately 270000. The mesh is shown in Figure B.1.

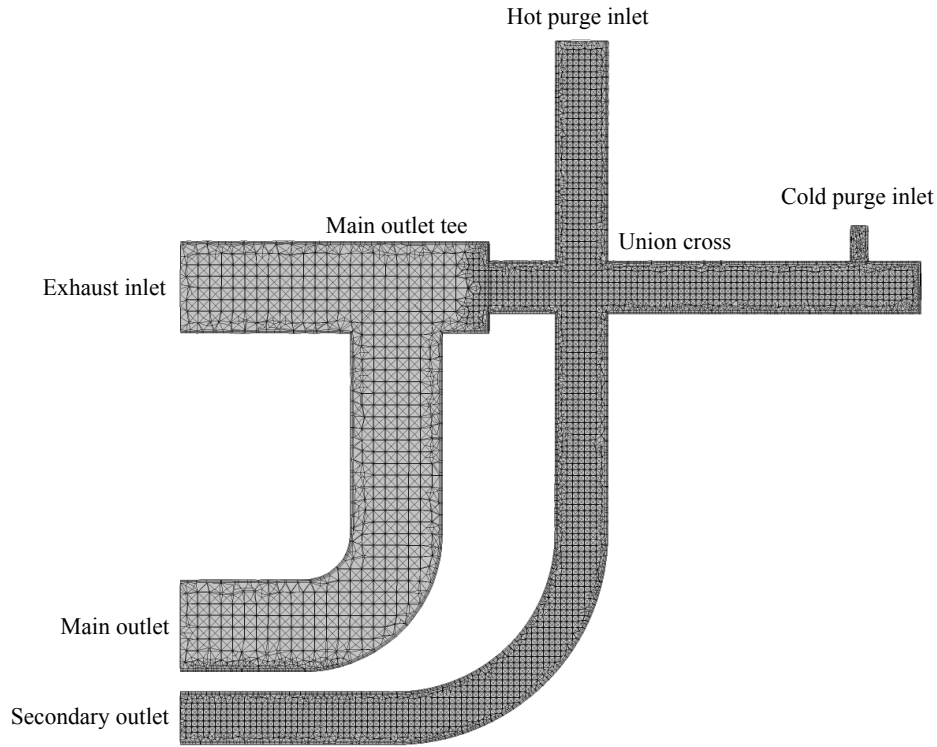


Figure B.1: Numerical mesh for the purging simulations.

B.2 Results

The results of the CFD simulations are evaluated by streamlines evolving from all three inlets in Figure B.2. The streamlines show the exhaust flow (red), the hot purge (orange), and the cold purge (blue). The exhaust mass flow of 25 kg/h was chosen for a high steam dilution ratio case, which was assumed as more critical regarding hot gas penetration in the cold purge section. The purge flow at both purging inlets was set to 1001/h. The streamlines show that most of the steam was guided from the tee to the main outlet. The rest flowed into the union cross and was guided into the secondary outlet, as intended. None of the steam got into the cold purge section. However, a stagnation effect could be observed for the cold purge flow, and a small portion of hot purge entered the cold purge section.

Thus, the purge flow was increased in two steps, as shown in Figure B.3. The mass flow rates were increased to 200 and 3001/h at both purge inlets in the first step. In the second step, the cold purge was further increased by 1001/h, while the hot one remained the same. The exhaust mass flow rate was kept constant in the three cases. Figure B.3 shows that the desired flow behavior was reached in all cases. Neither did the exhaust penetrate from the union cross into the cold purge section, nor did the N_2 contaminate the exhaust upstream of the union cross. At the highest purge rate, the N_2 purge of both inlets showed no stagnation in the union cross.

Figure B.4 shows the same three cases of purging but with only 10 kg/h exhaust gas. In none of the cases, any dilution effects could be observed. Moreover, the flow did not differ significantly with different purging flow rates.

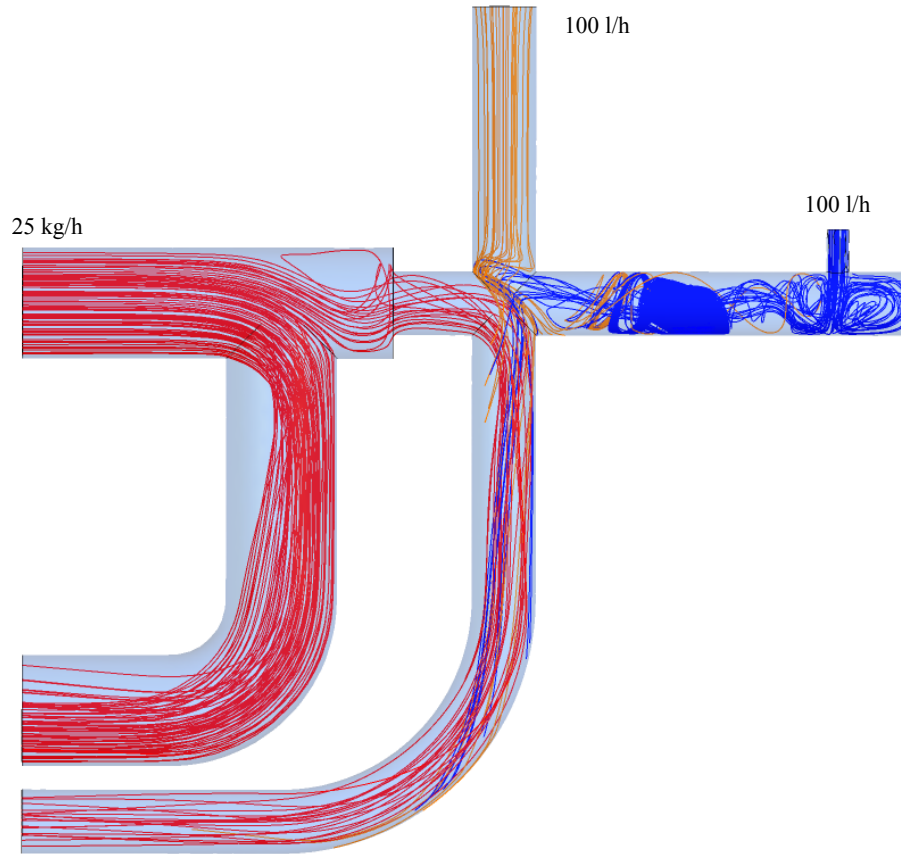


Figure B.2: Simulated flow in the TDLAS measurement section. The streamlines show the exhaust flow (red), the hot purge (orange) and the cold purge (blue).

Concluding this numerical pre-study, the purging levels from all the three cases seem to be viable for the measurements. Nevertheless, the highest purging flow was selected for the measurements to account for uncertainties.

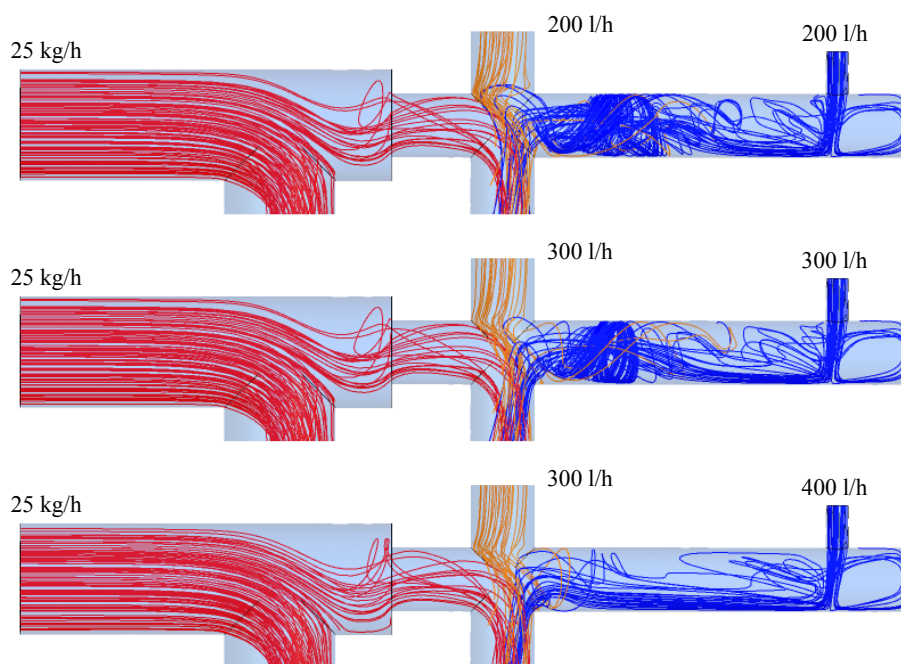


Figure B.3: Streamlines of the purge flow. The exhaust mass flow is kept constant at 25 kg/h while the purge flows are varied.

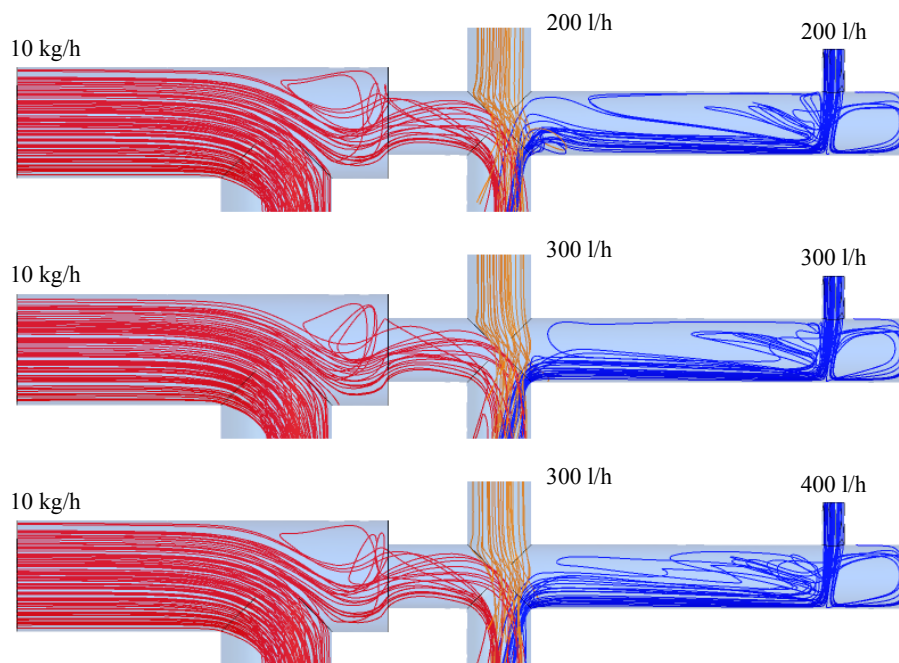


Figure B.4: Streamlines of the purge flow. The exhaust mass flow is kept constant at 10 kg/h while the purge flows are varied.

APPENDIX C

Additional Figures

In order to increase the readability of Chapter 5, some plots were shifted to the appendix. They are shown in the following figures.

C.1 TDLAS-based Combustion Efficiency Measurements

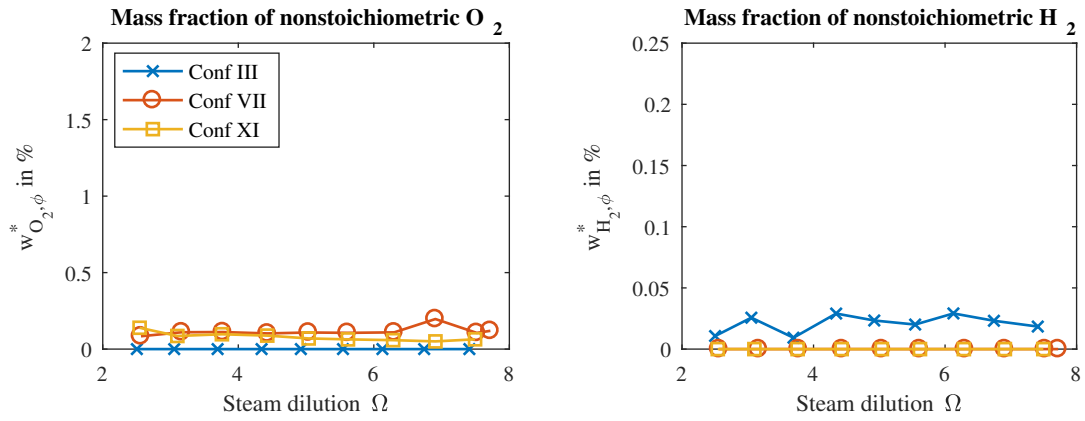


Figure C.1: Deviation from the stoichiometric ratio in mass fractions for burner at 30 kW thermal power. Left: O₂ measured by the Coriolis mass meters and Eq. 5.12. Right: H₂ measured by the Coriolis mass meters and Eq. 5.13.

C.2 TDLAS- and Lambda Sensor-based Combustion Efficiency Measurements

The mass fraction of the unburned H₂ was left out in the main text because it equals the H₂ fraction in the exhaust, shown in Fig. 5.20 (right), due to the negligible deviation from stoichiometric conditions. For completeness, it is shown here in Fig. C.2. The combustion efficiencies for 20 and 30 kW were shown in Sec. 5.5. The figures C.3 and C.4 show the concentrations of the 20 kW case. They are similar to the 30 kW case, which was shown exemplarily in Fig. 5.20 and 5.21.

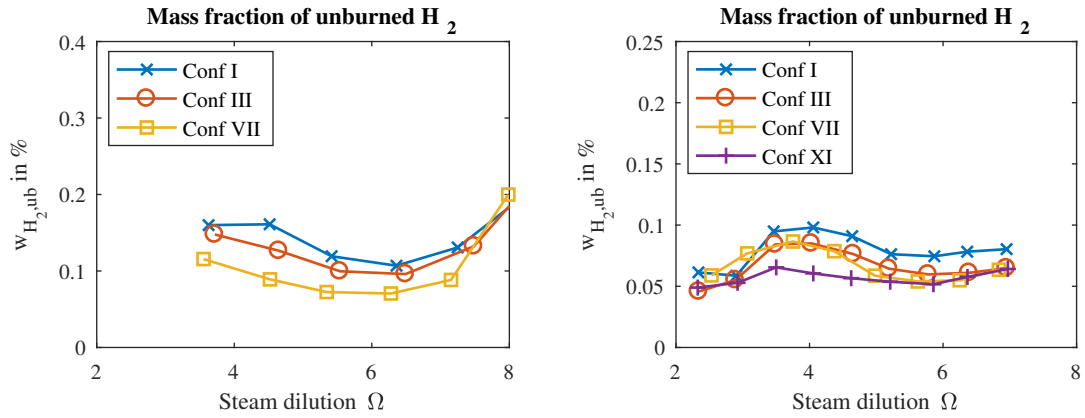


Figure C.2: Mass fraction of unburned H_2 for 20 kW (left) and 30 kW (right) calculated by Eq. 5.8.

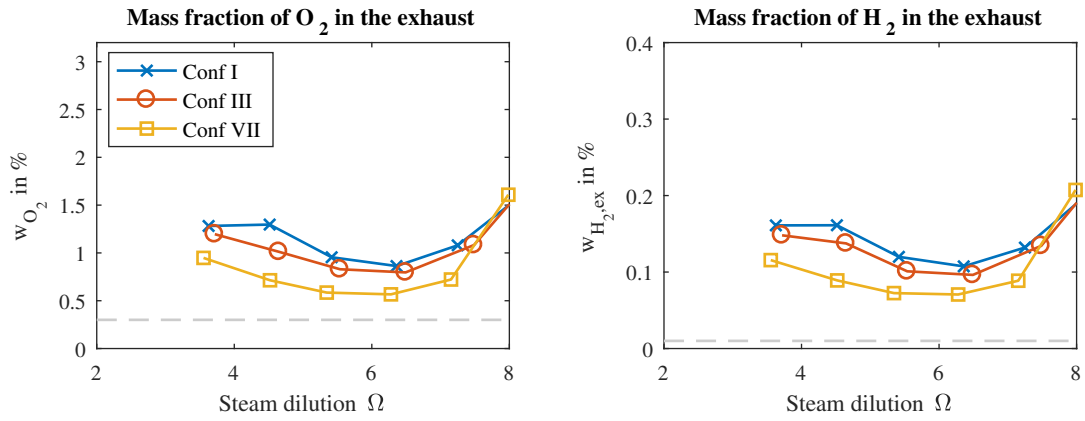


Figure C.3: Residual gas mass fractions in the exhaust at 20 kW thermal power. Left: O_2 measured by the TDLAS sensor. Right: H_2 calculated by Eq. 5.11.

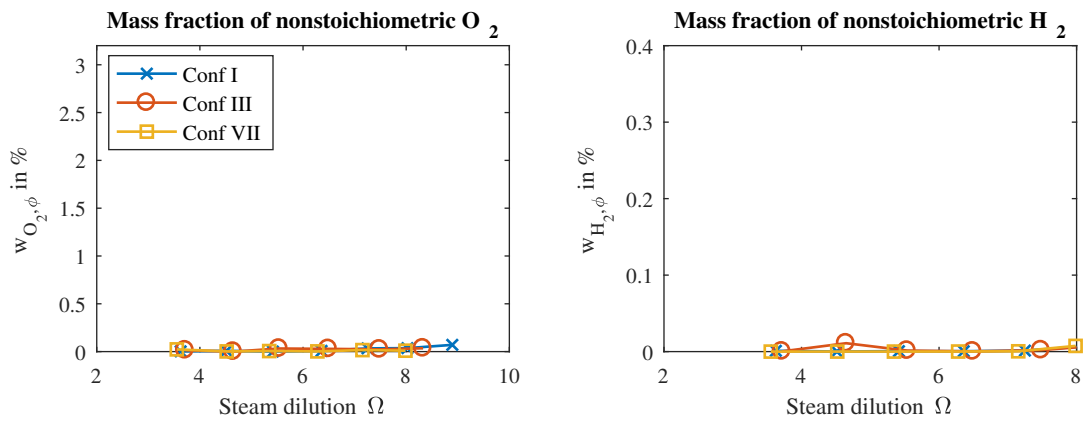


Figure C.4: Deviation from the stoichiometric ratio in mass fractions for burner at 20 kW thermal power. Left: O_2 measured by the LS and Eq. 5.4. Right: H_2 measured by the LS and Eq. 5.5.

APPENDIX D

Uncertainty Quantification of the Combustion Efficiency Measurements

The following section presents the measurement uncertainties of the LS and TDLAS measurements. In general, the uncertainty quantification follows the JCGM 100:2008 Guide to the expression of uncertainty in measurement [23]. Besides, the error measurement model by Fuller [64] was used to incorporate the calculated errors into the uncertainties of the LS models.

D.1 Error Propagation

Since the mass fractions were not measured directly, the uncertainties must be estimated for the entire measurement chain using the law of propagation of uncertainty [23]. For the output quantity $y = f(x_1, x_2, \dots, x_N)$ of the inputs x_i , it reads

$$\sigma^2(y) = \sum_{i=1}^N \left(\frac{\partial f}{\partial x_i} \right)^2 \sigma^2(x_i) + 2 \sum_{i=1}^{N-1} \sum_{j=i+1}^N \frac{\partial f}{\partial x_i} \frac{\partial f}{\partial x_j} \sigma(x_i, x_j) \quad (\text{D.1})$$

with $\sigma^2(x_i)$ and $\sigma(x_i, x_j)$ being the variances respectively covariances of the input quantities.

D.2 Uncertainty for the TDLAS-based measurements

The manufacturer reported the uncertainty of the TDLAS sensor as 0.1 %_v of the measurement value. The lower detection limit was 0.1 %_v as well. However, the signal required a correction for the measured temperature of the exhaust gas. By using Eq. (D.1) with an uncertainty of the thermocouple of 0.75 %, the uncertainty increases to 0.11 %_v, respectively $\sigma_{w_{\text{O}_2}} = 0.19 \text{ \%}_m$ (at $T_{\text{TDLAS}} = 773 \text{ K}$) in terms of mass fractions.

The TDLAS-based combustion efficiency (without LS) was calculated by Eq. (5.16). The input quantities were the reactant and exhaust mass flows, as well as the mass fraction of unburned H_2 . The uncertainties of the mass flows are plotted in Fig. 5.19. The uncertainties of the mass fractions were calculated from the uncertainty of the TDLAS sensor using Eq. (5.12)-(5.15) and the law of error propagation (Eq. (D.1)). Table D.1 displays them for a reference case with 30 kW thermal power and $\Omega = 5$. The uncertainties were evaluated for three different equivalence ratios ($\phi = 0.98, 1, 1.02$) because the calculation of the mass fractions changes if the flame is rich, lean or stoichiometric (see Eq. (5.17)) and all three cases are conceivable. In the case of the lean flame, the non-stoichiometric H_2 fraction is zero by definition, and so is the uncertainty. In return, the uncertainty of $w_{\text{O}_2, \phi}$ is relatively high, which reflects in an increased $w_{\text{H}_2, \text{ub}}$ uncertainty for the lean flame. This parameter

has a strong influence on the combustion efficiency. Figure D.1 shows that the uncertainty of η^* is nearly twice as high for the lean flame. However, the equivalence ratio was determined by the mass meters and their uncertainties in the measurements. Consequently, the decision itself, if the flame was rich or lean, was highly uncertain for operation close to stoichiometric conditions. The standard deviation of the equivalence ratio around $\phi = 1$ is plotted in Fig. 5.19 (right). If the operating point is within this range, the higher uncertainty of $\sigma_{\eta^*} \approx 2.9\%$ should be used to estimate the error. σ_{η^*} is nearly constant for the relevant range of unburned H_2 fraction.

Table D.1: TDLAS-based approach: uncertainties of the measured mass fraction for a lean, a stoichiometric and a rich flame. The input parameter are $P_{th} = 30 \text{ kW}$, $\Omega = 5$ and $T_{TDLAS} = 773 \text{ K}$.

σ	$\phi=0.98$	$\phi=1$	$\phi=1.02$	
$w_{O_2,\phi}^*$	0.38	0	0	% _m
$w_{H_2,\phi}^*$	0	0	0.38	% _m
$w_{O_2,ub}^*$	0.42	0.19	0.19	% _m
$w_{H_2,ub}^*$	0.053	0.024	0.024	% _m
$w_{O_2,ex}$	0.19	0.19	0.19	% _m
$w_{H_2,ex}^*$	0.053	0.024	0.38	% _m

Please note that the high uncertainty of the combustion efficiency was caused by the high uncertainty of the H_2 Coriolis mass meter (see Fig. 5.19). If one assumes a typical uncertainty of 0.5 % of the measured value, the uncertainty of the combustion efficiency would be below $\sigma_{\eta^*} = 1.5\%$ for both the rich and the lean flame.

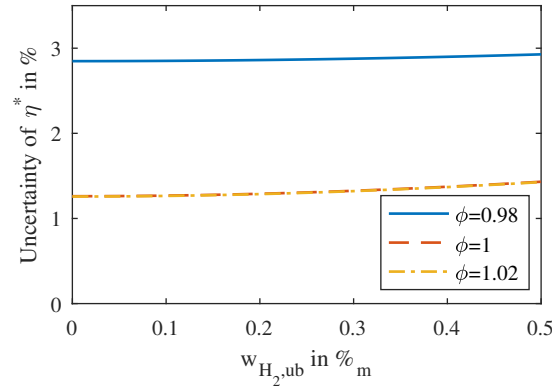


Figure D.1: Uncertainties of the TDLAS-based combustion efficiency for a lean, a stoichiometric and a rich flame. The input parameter are $P_{th} = 30 \text{ kW}$, $\Omega = 5$ and $T_{TDLAS} = 773 \text{ K}$.

D.3 Uncertainty of the TDLAS- and LS-based measurements

In the TDLAS- and LS-based approach, the equivalence ratio was not determined by the Coriolis mass meters but by the LS, which was calibrated before usage, as described in Sec. 5.3.1. In order to evaluate the accuracy of this calibration, a measurement error model was used.

D.3.1 Measurement Error Model

The models used to evaluate the combustion efficiency in this work can be rewritten into a linear regression. Therefore, the measurement error model with known measurement variance by Fuller [64] could be used to estimate their uncertainties. Let the measured values be

$$X_t = x_t + u_t, Y_t = c_0 + c_1 x_t + e_t, \quad (\text{D.2})$$

with the uncertainty of the measurement σ_{uu} . From the calibration samples, the covariances were calculated with

$$m_{XY} = \frac{1}{n-1} \sum_{t=1}^n (X_t - \bar{X}) (Y_t - \bar{Y}). \quad (\text{D.3})$$

The regression coefficients were estimated to

$$c_1 = \frac{m_{XY}}{m_{XX} - \sigma_{uu}}, c_0 = \bar{y} - c_1 \bar{X}, \quad (\text{D.4})$$

in the case of non-negative $\sigma_{xx} = m_{XX} - \sigma_{uu}$ and $\sigma_{ee} = m_{YY} - c_1 m_{XY}$. Else, the estimators must be adjusted to

$$c_1 = \frac{m_{YY}}{m_{XY}}, \quad (\text{D.5})$$

$\sigma_{xx} = m_{XY}/c_1$ and $\sigma_{ee} = 0$. The variances of the model coefficients were estimated by

$$\sigma^2(c_1) = \frac{1}{(n-1)\sigma_{xx}^2} (m_{XX}s_{vv} + c_1^2\sigma_{uu}^2), \quad (\text{D.6})$$

$$\sigma^2(c_2) = \bar{X}^2 \sigma^2(c_1) + \frac{s_{vv}}{n}, \quad (\text{D.7})$$

$$\sigma(c_1, c_0) = -\bar{X} \sigma^2(c_1) \quad (\text{D.8})$$

with

$$s_{vv} = \frac{1}{n-2} \sum_{t=1}^n (Y_t - \bar{Y} - (X_t - \bar{X}) c_1)^2. \quad (\text{D.9})$$

D.3.2 Uncertainty of the LS Model

As described before, the calibration of the LS was done using a fine metering valve. Its uncertainty was estimated using Eq. (D.1)-(D.8) to $\sigma(\dot{m}_{\text{O}_2, \text{set}}) = 0.0039 \text{ kg/h}$. With the uncertainty of the steam mass flow of $\sigma(\dot{m}_{\text{H}_2\text{O}, \text{set}}) = 0.25 \text{ kg/h}$, the uncertainty of the set mass fraction in the calibration was estimated by Eq. (D.1) to $\sigma(w_{\text{O}_2, \text{set}}) < 0.011\%_m$ at calibration conditions. The measurement error model was applied using these values as input to determine the coefficients for the O_2 model (Eq. (5.2)) and their uncertainties. However, the measurement error model failed due to the exponential nature of the problem and was thus modified by using $\sigma_{uu} = 0 \text{ V/K}$ in Eq. (D.5) and Eq. (D.6). Nevertheless, the previous measurement errors were still incorporated by Eq. (D.10), which estimated the

overall uncertainty for the LS measurements:

$$\begin{aligned} \sigma^2(w_{\text{O}_2,\phi}) = & \left(\frac{\partial w_{\text{O}_2}}{\partial c_1} \right)^2 \sigma^2(c_1) + \left(\frac{\partial w_{\text{O}_2}}{\partial c_2} \right)^2 \sigma^2(c_2) \\ & + 2 \frac{\partial w_{\text{O}_2}}{\partial c_1} \frac{\partial w_{\text{O}_2}}{\partial c_2} \sigma(c_1, c_2) \\ & + \left(\frac{\partial w_{\text{O}_2}}{\partial U_{\text{LS}}} \right)^2 \sigma^2(U_{\text{LS}}) + \left(\frac{\partial w_{\text{O}_2}}{\partial T_{\text{LS}}} \right)^2 \sigma^2(T_{\text{LS}}). \end{aligned} \quad (\text{D.10})$$

with $\sigma(U_{\text{LS}}) = 1.8 \text{ mV}$ and $\sigma(T_{\text{LS}}) = 0.5 \text{ K}$. The model coefficients $c_{1,\text{O}_2} = -5.51 \text{ e}+4$ and $c_{2,\text{O}_2} = -0.34$ were calculated using the prescribed measurement error model with $X_t = U_t/T_t$ and $Y_t = \log(w_{\text{O}_2,t})$ being the measured calibration values. Figure D.2 gives the result of the uncertainty estimation. The uncertainty increases as the deviation from $\phi = 1$ rises. However, as the LS has its maximum sensitivity in the stoichiometric point, $\phi = 1$ can be set very precisely. The maximum value of $w_{\text{O}_2,\phi}$, observed in the results of Sec. 5.5, was only about $0.07 \%_{\text{m}}$. The corresponding uncertainty is $\sigma_{w_{\text{O}_2,\phi}} = 0.01 \%_{\text{m}}$.

The same approach was used for the LS H_2 model (Eq. (5.5)) with $X_t = U_t/T_t$ and $Y_t = \log(w_{\text{H}_2,t})$. The uncertainty of $w_{\text{H}_2,\phi}$ was estimated using the error propagation (Eq. (D.1)) in the same form as for the O_2 model (Eq. (D.10)). The model coefficients were $c_{1,\text{H}_2} = 2.134 \text{ e}+4 \text{ V/K}$ and $c_{2,\text{H}_2} = -23.92$. This led to the uncertainties in Figure D.2. The maximum value of $w_{\text{H}_2,\phi}$, observed in the measurement of this thesis, was $0.01 \%_{\text{m}}$. The corresponding uncertainty is $\sigma_{w_{\text{H}_2,\phi}} = 0.0008 \%_{\text{m}}$.

D.3.3 Uncertainty of the Combustion Efficiency

The evaluation of the combustion efficiency's uncertainty followed the same approach as in Sec. D.2. It was calculated by

$$\sigma_\eta^2 = \left(\frac{\partial \eta}{\partial w_{\text{H}_2,\text{ub}}} \right)^2 \sigma_{w_{\text{H}_2,\text{ub}}}^2 + \left(\frac{\partial \eta}{\partial \dot{m}_{\text{ex}}} \right)^2 \sigma_{\dot{m}_{\text{ex}}}^2 + \left(\frac{\partial \eta}{\partial \dot{m}_{\text{H}_2}} \right)^2 \sigma_{\dot{m}_{\text{H}_2}}^2. \quad (\text{D.11})$$

However, the uncertainties of the nonstoichiometric mass fractions $\sigma_{w_{\text{O}_2,\phi}}$ were not calculated from the uncertainties of the mass meters but from the model error in Eq. (D.10). It influenced the first term in Eq. (D.11), which has been the largest one. Compared to the

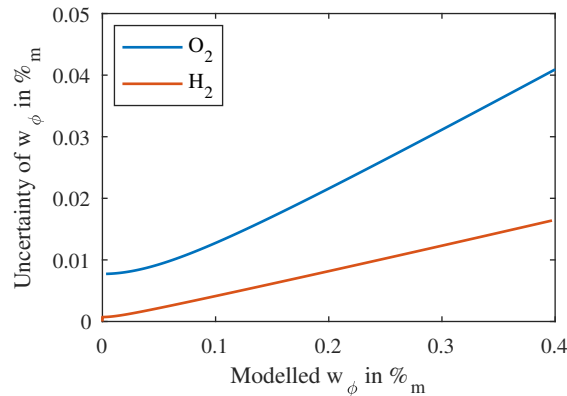


Figure D.2: Uncertainty of the mass fraction of the residual gases' non-stoichiometric parts, which are measured by the LS.

TDLAS-based approach in Fig. D.1, the influence of the deviation from $\phi = 1$ was much lower, as shown in Fig. D.3. At $w_{\text{H}_2,\text{ub}} = 0.1\%_{\text{m}}$, which was the highest observed unburned fraction, the uncertainty of η is about 1.28%.

Table D.2: TDLAS- and LS-based approach: uncertainties of the measured mass fraction for a lean, a stoichiometric and a rich flame. The input parameter are $P_{\text{th}} = 30\text{ kW}$, $\Omega = 5$ and $T_{\text{TDLAS}} = 773$.

σ	$\phi=0.98$	$\phi=1$	$\phi=1.02$	
$w_{\text{O}_2,\phi}$	0.031	0.0077	0	$\%_{\text{m}}$
$w_{\text{H}_2,\phi}$	0	0.00072	0.0017	$\%_{\text{m}}$
$w_{\text{O}_2,\text{ub}}$	0.19	0.19	0.19	$\%_{\text{m}}$
$w_{\text{H}_2,\text{ub}}$	0.024	0.024	0.024	$\%_{\text{m}}$
$w_{\text{O}_2,\text{ex}}$	0.19	0.19	0.19	$\%_{\text{m}}$
$w_{\text{H}_2,\text{ex}}$	0.024	0.024	0.024	$\%_{\text{m}}$

Please note that the uncertainty of the combustion efficiency increased due to the high uncertainty of the H_2 Coriolis mass meter (see Fig. 5.19), which influenced the third term of Eq. (D.11). If one assumes a typical uncertainty of 0.5% of the measured value, the uncertainty of the combustion efficiency would be slightly lower, especially for sizable unburned mass fractions.

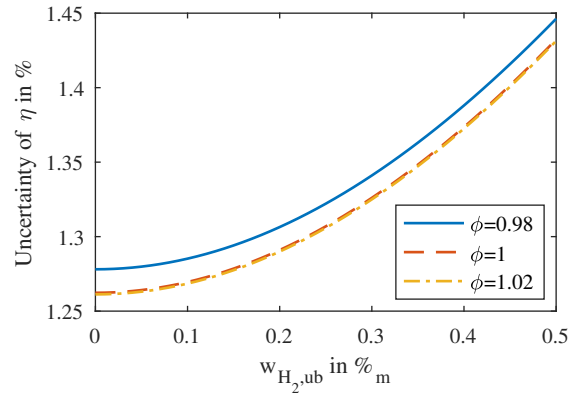


Figure D.3: Uncertainties of the TDLAS- and LS-based combustion efficiency for a lean, a stoichiometric and a rich flame. The input parameter are $P_{\text{th}} = 30\text{ kW}$, $\Omega = 5$ and $T_{\text{TDLAS}} = 773\text{ K}$.

APPENDIX E

Publications Associated with the Thesis

Parts of this thesis have been published previously by the author. The following list gives an overview of these publications. The concepts, ideas, and methods expressed in this thesis have been developed by the author. Scientific guidance have been provided by Panagiotis Stathopoulos and Christian Oliver Paschereit. The students Josepf Seefeldt, Max Kossatz, and Simeon Dybe contributed to the thesis in the scope of their bachelor or master thesis. Janis Mörsdorf further assisted in the data acquisition.

- Schimek, S. and Stathopoulos, P. and **Tanneberger, T.** and Paschereit, C. O. (2015), Blue Combustion: Stoichiometric Hydrogen-Oxygen Combustion Under Humidified Conditions. In Volume 4B: Combustion, Fuels and Emissions. American Society of Mechanical Engineers.
doi: 10.1115/gt2015-43149.
- **Tanneberger, T.** and Schimek, S. and Kossatz, M. and Paschereit, C. O. and Stathopoulos, P. (2017), Development of a Swirl-Stabilized H₂/O₂ Combustion System under Humidified Conditions. In Digital Proceedings of the 8th European Combustion Meeting (ECM 2017).
uri: 10.14279/depositonce-5948.
- **Tanneberger, T.** and Schimek, S. and Paschereit, C. O. and Stathopoulos, P., Combustion efficiency measurements and burner characterization in a hydrogen-oxyfuel combustor. International Journal of Hydrogen Energy, Bd. 44(2019)(56).
doi: 10.1016/j.ijhydene.2019.05.055.
- **Tanneberger, T.** and Schimek, S. and Paschereit, C. O. and Stathopoulos, P., Efficiency Measurement Approach for a Hydrogen Oxyfuel Combustor. Journal of Engineering for Gas Turbines and Power, Bd. 141(2019)(10).
doi: 10.1115/1.4044779.
- Dybe, S. and **Tanneberger, T.** and Stathopoulos, P., Second Law Analysis of an Energy Storage System Consisting of an Electrolysis Plant and the Graz Cycle With Internal H₂/O₂ Combustion. Journal of Engineering for Gas Turbines and Power, Bd. 141(2019)(11).
doi: 10.1115/1.4044451.
- **Tanneberger, T.** and Stathopoulos, P. (2020), Heat Transfer Measurements in a Hydrogen-Oxyfuel Combustor. Experimental Heat Transfer.
doi: 10.1080/08916152.2020.1798564.