Application of lean premixed prevaporized burner for renewable fuels

PhD Thesis by
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Supervisor
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Abstract

Renewable liquid fuels play an essential role in our efforts to achieve a sustainable economy. In this thesis, the operation of a lean premixed prevaporized, LPP, burner is investigated in detail. Firstly, spray characteristics were measured and evaluated, emphasizing droplet dynamics, global droplet sizes, and size distribution functions. Secondly, evaporation of fossil and renewable fuels were calculated and compared in the case of the investigated LPP burner. Next, combustion tests were carried out mainly comparing diesel and rape-seed oils from blowout and pollutant emission point of view. Diffusers of various half cone angles were analyzed with a conclusion that they may significantly enhance the blowout stability. Finally, flame emission spectrometry was used to investigate especially the turbulent flame in the bistable operating region while it is a highly potential tool to accomplish online flame control.

Keywords: Airblast atomization, droplet evaporation, pollutant emission, flame stability, rapeseed oil, flame emission spectrometry
Declaration

Undersigned, Viktor Józsa, author of this Thesis, hereby, I declare that I prepared this Thesis myself, and I used only the referred sources. All parts of the Thesis, which are either cited verbatim or with the same content but rephrased as in the original source, are referred unambiguously and with providing the sources used.

Budapest, June 6, 2016

Viktor Józsa
Acknowledgments

Hereby, I would like to express my gratitude to all who helped me to finish my PhD studies. My family, my colleagues, and friends. Without their support, it would have been impossible.

Among my colleagues, I would like to highlight my supervisor, Krisztiján Sztankó. I really appreciate his personal support and advices – even out of the technical details. The collaboration with his another PhD Student, Attila Kun-Balog, contributed significantly to this work in numerous measurements, evaluation, and discussion.

The helpfulness and the support of Gyula Gróf, the Head of Department, was also outstanding. His technical advices, like the use of a spectrometer, greatly contributed to the scientific value of the present thesis. Moreover, he encouraged and supported me to participate in world-class conferences where I could meet with the widely recognized experts of combustion technology, such as Assad Masri, Sébastien Candel, and Mário Costa. I appreciate the support of Tamás Fülöp, who encouraged me to write my thesis using LATEX. I also acknowledge the constructive comments of my colleagues and the pre-review to Csaba Hős and Ferenc Lezsovits.

I would like to give my special thanks to Jan Jedelský, Matouš Zaremba, Milan Malý, and András Urbán, who carried out the PDA measurements of our burner at Brno University of Technology. Their insights regarding the analysis of atomization were really appreciated. It was a pleasure to work with them. The funding of the Visegrád 3–111–0027 Strategic Grant, V4 Green energy platform is acknowledged.

I am grateful for the support of my students who were involved in various aspects of the present research. I am very proud that many of them found a personal challenge behind the actual project and also participated with their topics in the Scientific Students’ Associations. They are András Urbán, Áron Tóth, Boglárka Kurucz, Dávid Csemány, Gyöngyvér Hidegh, and Olivér Wágner.

Finally, I would like to express my gratitude to my girlfriend, Ági, for her love and support to write this dissertation.
Collaborations

While all the chapters – excluding the literature-based introduction – contains a cooperative work with researchers and students beside my supervisor, this part summarizes and details the contribution of the involved people to this PhD Thesis. Consequently, no thesis was stated based on these parts in order to avoid any possible conflicts.

The measurement of Chapter 3 was performed at the Brno University of Technology through a joint project in July 2015, funded by the Visegrád 3–111–0027 Strategic Grant, V4 Green energy platform. Here, Section 3.2 is mainly the work of Matouš Zaremba, MSc. The reason of the inclusion of this single section is that some later phenomena were explained by using the results of droplet dynamics. Jan Jedelský, PhD, Milan Malý, BSc, András Urbán, BSc were also involved in this research, but no conflict of interest was founded, they approved that the remaining part of the chapter as my own work, including the 1st Thesis. Historically, the need for an appropriate SMD estimation, which preceded the investigation of atomization, was based on the calculation of evaporation, discussed in Chapter 4. The basis of this chapter is a publication with Dávid Csemány, BSc.

The measurements of Chapters 5 and 6 were carried out in the György Jendrassik Laboratory of Department of Energy Engineering. Next to Beside Krisztián Sztankó, PhD, my supervisor, his other PhD Student, Attila Kun-Balog, MSc participated in almost all the measurements and evaluation. As we worked together in numerous investigations, we have agreed which part corresponds to who at the beginning, regarding our PhD studies. Therefore, no conflict of interest was stated, and our theses are well separated. Gyöngyvér Hidegh, BSc also participated in several measurements, which resulted in a publication. She approved that the results that are connected to the 2nd Thesis are based on my work. Áron Tóth, BSc also helped in few measurements.

All the co-authors of the theses-related publications signed a concession statement, as it is requested. These documents are attached to the submitted PhD thesis.
## Contents

Abstract ............................................................................... ii
Declaration ........................................................................ iii
Acknowledgments ................................................................ iv
Collaborations ..................................................................... v
List of Figures ....................................................................... viii
List of Tables ......................................................................... xi
Nomenclature ......................................................................... xii

1 Objective and outline .......................................................... 1

2 Introduction ........................................................................ 5
  2.1 Combustion of liquid fuels .............................................. 5
  2.2 Atomization and spray characterization ......................... 6
    2.2.1 Estimation of the mean droplet diameter ................. 8
    2.2.2 Droplet size distribution functions ......................... 11
    2.2.3 Droplet stability .................................................. 11
  2.3 Evaporation of droplets ................................................ 12
  2.4 Combustion in modern burners ....................................... 15
    2.4.1 Flow patterns ..................................................... 15
    2.4.2 Combustion stability ............................................ 18
    2.4.3 Pollutant emissions ............................................. 20
  2.5 Flame diagnostic techniques ......................................... 23
    2.5.1 Flame emission spectrometry ............................... 24

3 Atomization ......................................................................... 26
  3.1 Phase Doppler Anemometry measurement setup ............. 27
  3.2 Droplet dynamics ....................................................... 29
  3.3 Global spray characterization ....................................... 33
  3.4 Validation of SMD estimating formulas ......................... 34
  3.5 Droplet size distribution .............................................. 36
  3.6 Summary of the chapter ................................................ 39
4 Evaporation
  4.1 Calculation of the evaporation number . . . . . . . . . . . . . . . . . . . . 41
  4.2 Fuel and combustion air preheating . . . . . . . . . . . . . . . . . . . . . 45
  4.3 Summary of the chapter . . . . . . . . . . . . . . . . . . . . . . . . . . . . 47

5 Combustion in a LPP burner
  5.1 The measurement system . . . . . . . . . . . . . . . . . . . . . . . . . . . 48
  5.2 Experimental stability analysis . . . . . . . . . . . . . . . . . . . . . . . . 50
  5.3 Flame stabilizing solutions . . . . . . . . . . . . . . . . . . . . . . . . . . 52
    5.3.1 Analysis of various burner configurations . . . . . . . . . . . . . . . 54
    5.3.2 Application of diffusers . . . . . . . . . . . . . . . . . . . . . . . . . 57
  5.4 Pollutant emission analysis . . . . . . . . . . . . . . . . . . . . . . . . . . 60
    5.4.1 Comparison of rapeseed oil and diesel oil combustion . . . . . . . . 60
    5.4.2 Effect of the diffuser, the confinement, and the elevated combustion
         power on pollutant emissions . . . . . . . . . . . . . . . . . . . . . . . . . 64
  5.5 Summary of the chapter . . . . . . . . . . . . . . . . . . . . . . . . . . . . 67

6 Spectrometry
  6.1 The spectrometer . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 68
  6.2 Evaluation methods . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 69
  6.3 Spatial FES measurement . . . . . . . . . . . . . . . . . . . . . . . . . . . 71
  6.4 FES of rapeseed oil . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 74
  6.5 Summary of the chapter . . . . . . . . . . . . . . . . . . . . . . . . . . . . 77

Conclusions and future directions

Theses

References

A Uncertainty and data sheet of measuring devices

B Estimation of the swirl number

C Detailed fuel properties for calculating evaporation

D Blowout characteristics
List of Figures

1.1 Primary mechanisms and their relation of premixed combustion. Thick borders indicate the investigated phenomena. ............................................. 2
1.2 Flow pattern of a preswirled burner (left) and section of a gas turbine combustion chamber (right). .......................................................... 2
1.3 Blowout during a knife-edge turn, performed by an SR-71 Blackbird. ... 4
2.1 Nukiyama and Tanasawa (left) and Lefebvre and Miller atomizer (right). 6
2.2 Classification of disintegration modes. ............................................. 8
2.3 Theoretical model for the evaporation of a single droplet. .................. 13
2.4 IRZ and the ORZ at a) 10 kW and b) 35 kW combustion power, indicating the absolute velocity (left) and the tangential velocity (right). ............ 16
2.5 Straight (left) and V-shaped flame (right). ....................................... 17
2.6 Comparison of the flow field with and without a diffuser. .................. 17
2.7 Influence of atomizing pressure on flame stability. .......................... 19
2.8 Ignition loop of a burner. ............................................................... 19
2.9 Effect of equivalence ratio on pollutant emissions and temperature. ....... 21
3.1 The LPP burner without the mixing tube. ......................................... 26
3.2 The atomization test rig. ................................................................. 27
3.3 PDA measurement system. .............................................................. 28
3.4 Size-velocity correlations at \( p_g = 0.9 \) bar and various axial distances: a) \( z = 10 \) mm, b) \( z = 15 \) mm, c) \( z = 26.7 \) mm, and d) \( z = 50 \) mm. All results were obtained at \( r = 0 \) mm. ......................................................... 29
3.5 Size-velocity correlations at \( z = 50 \) m and various atomizing pressures: a) \( p_g = 0.3 \) bar, b) \( p_g = 0.9 \) bar, c) \( p_g = 2.1 \) bar, and d) \( p_g = 3.1 \) bar. All results were obtained at \( r = 0 \) mm. ......................................................... 30
3.6 Radial profiles of TKE, MKE, axial and radial velocity profiles. All of these characteristics are shown for liquid and gas phase in the spray and for: a) \( z = 10 \) mm, \( p_g = 0.9 \) bar, b) \( z = 15 \) mm, \( p_g = 0.9 \) bar, c) \( z = 26.7 \) mm, \( p_g = 0.9 \) bar, d) \( z = 50 \) mm, \( p_g = 0.9 \) bar, e) \( z = 50 \) mm, \( p_g = 0.3 \) bar, f) \( z = 50 \) mm, \( p_g = 0.9 \) bar, g) \( z = 50 \) mm, \( p_g = 2.1 \) bar, h) \( z = 50 \) mm, \( p_g = 3.1 \) bar. Note the different radial scale in a) and b). .............................. 31
3.7 Size-velocity correlation at \( z = 50 \) mm and \( r = 0 \) mm a) \( p_g = 0.3 \) bar and b) \( p_g = 0.3, 0.9, 2.1, \) and 3.1 bar. ......................................................... 32
3.8 Radial SMD distribution of the spray at a) \( p_g = 0.3 \) bar, b) \( p_g = 0.9 \) bar, c) \( p_g = 2.1 \) bar, and d) \( p_g = 3.1 \) bar. ......................................................... 33
3.9 Comparison of the a) calculated stable droplet sizes with b) SMD at \( z = 50 \) mm. ................................................................. 33
3.10 ISMD of the spray at various axial distances. .................................. 34
3.11 Fitted SMD estimations at $z = 50$ mm based on Eqs. 2.6–2.11. 35
3.12 Average parameters of the a) $\Gamma$, RR, and b) NT PDFs at $z = 50$ mm. 37
3.13 Droplet size distributions at various atomizing pressure, using $\Gamma$ PDF. 37
3.14 Averaged coefficient of determination of the fits. 38
3.15 Integral of the averaged PDFs. 38

4.1 The investigated LPP burner. 40
4.2 Test rig for thrust measurement. 41
4.3 Comparison of the measured and calculated mass flow rates through adiabatic expansion. 42
4.4 SMD of different fuels at various atomizing pressures. 43
4.5 Ratio of residence and evaporation time for different fuels, $T_{s,0} = 21$ °C, $T_{s,\text{st}} = 700$ K. 45
4.6 Ratio of residence and evaporation time for rapeseed and soybean oils with fuel preheating, $T_{s,\text{st}} = 700$ K. 46
4.7 Ratio of residence and evaporation time for rapeseed and soybean oils with combustion air preheating, $T_{s,0} = 21$ °C. 46

5.1 The combustion test rig. 48
5.2 Swirl number of a) diesel oil and b) rapeseed oil combustion. 51
5.3 Sequences from the bistable state of the flame at $p_g = 0.7$ bar. The time step between the frames is 1/30 s. 52
5.4 Detached (left) and attached flow (right) at $p_g = 0.7$ and 0.35 bar, respectively. A 45° diffuser was applied. 54
5.5 a) original configuration, b) with a 45° diffuser, c) with a 17° diffuser, and d) with a 17° converging nozzle. 54
5.6 Blowout stability of the investigated configurations. 55
5.7 Flame of the converging nozzle. 56
5.8 Sequences from blowout in the case of the original burner, $p_g = 0.7$ bar. 56
5.9 Sequences from blowout with a 45° diffuser, $p_g = 0.7$ bar. 57
5.10 Cross section of the diffusers. $\theta$ is a parameter, indicating the half cone angle. 58
5.11 Blowout stability of the diffusers. 58
5.12 Start of the bistable region. 59
5.13 Start of the fully developed V-shaped flame region. 59
5.14 Comparison of the stability limits of the original and the 15° diffuser configurations. 60
5.15 a) NO$_X$ and b) CO emission of diesel oil combustion. 61
5.16 a) NO$_X$ and b) CO emission of rapeseed oil combustion. 62
5.17 Overall emission of a) diesel oil and b) rapeseed oil combustion. 63
5.18 Straight (left) and V-shaped (middle) flame of the original burner and the V-shaped flame with the 45° diffuser (right). 64
5.19 Emissions of the original burner and with a 45° diffuser. 64
5.20 The burner lip with the confining tube. 65
5.21 Measured pollutant emission at the stack at 15% O$_2$ level. 66

6.1 The spectrometer. 68
6.2 Spectra of rapeseed oil and diesel oil combustion at $p_g = 1$bar and $\lambda = 1.15$. Note that only dark current correction was performed for better visibility. 69
6.3 Effect of $\lambda$ and $p_g$ on the chemiluminescence emission spectra through the appearance of soot. ....................................................... 70
6.4 Spectra of crude sunflower oil and diesel oil combustion at $p_g = 3$ bar and $\lambda = 1.15$. ................................................................. 71
6.5 Raw OH*, CH*, C*$_{2,516}$ spatial chemiluminescence distribution at a 0.35 bar (left), 0.7 bar (middle), and 1 bar (right) atomizing pressure. .... 72
6.6 Corrected OH*/CH*, OH*/C*$_{2,516}$, CH*/C*$_{2,516}$ chemiluminescence ratios along the axis. ............................................................. 73
6.7 Corrected chemiluminescence intensity ratios of OH*, CH*, C*$_{2,516}$ to C*$_{2,554}$ of rapeseed oil combustion. ........................................ 74
6.8 Corrected chemiluminescence intensity ratios of OH*/CH* of diesel oil (top) and rapeseed oil combustion (bottom). .................................. 75
6.9 Corrected chemiluminescence intensity ratios of OH*/C*$_{2,516}$ of diesel oil (top) and rapeseed oil combustion (bottom). .......................... 76
6.10 Corrected chemiluminescence intensity ratios of CH*/C*$_{2,516}$ of diesel oil (top) and rapeseed oil combustion (bottom). ....................... 77
A.1 The atomization test rig (detailed). .................................................. 95
A.2 The combustion test rig (detailed). .................................................... 97
B.1 The cross section of the swirl vanes. .................................................. 101
D.1 Blowout without stabilizer. Sequences from high speed video with inverted colors. The teal lines indicate the burner lip. ......................... 106
D.2 Blowout with a 45° diffuser. Sequences from high speed video with inverted colors. The teal lines indicate the burner lip. ......................... 107
List of Tables

2.1 Test conditions of the discussed SMD estimating formulas. .................. 10
3.1 Summary of the fitted equations of Fig. 3.11. ................................. 36
4.1 Probability of different fatty acids in the triglyceride of the investigated vegetable oils. ................................................................. 42
4.2 Relevant physical properties of the investigated fuels for SMD calculation. 43
4.3 Atomization and evaporation characteristics at different gauge pressures and 700 K combustion air temperature. .............................. 44
4.4 Temperature-dependent properties of rapeseed and soybean oils. ........ 45
5.1 Fuel properties. ........................................................................ 49
5.2 Measurement conditions of Sections 5.2, 5.4.1, and 6.4. .................... 53
5.3 Combustion of diesel oil and rapeseed oil with and without confinement and at 15 and 16.8 kW. ......................................................... 65
6.1 List of the considered chemiluminescence emitters and their respective higher and lower 5 nm wide bands for correction. ...................... 70
A.1 Accuracy of the monitored modules of the Testo 350 flue gas analyzer. ... 99
A.2 Calibration results of the Testo 350 flue gas analyzer. ......................... 100
C.1 Fuel properties at 21 °C. ............................................................. 103
C.2 Constants for Eq. C.2. ................................................................ 104
C.3 Constants of ethanol vapor pressure. ............................................. 104
Nomenclature

The tables below contain the notation, description, and the corresponding unit of variables used in the present work. However, rarely used variables and subscripts are introduced at their occurrences. All the used variables are in SI units.

**Latin letters**

<table>
<thead>
<tr>
<th>Notation</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a, b, g, q)</td>
<td>constants (in distribution functions)</td>
<td>-</td>
</tr>
<tr>
<td>(A, B, C, E, F, H, J, K)</td>
<td>constants (in SMD estimation)</td>
<td>-</td>
</tr>
<tr>
<td>(c)</td>
<td>speed of sound</td>
<td>m/s</td>
</tr>
<tr>
<td>(c_p)</td>
<td>specific heat capacity at constant pressure</td>
<td>J/kgK</td>
</tr>
<tr>
<td>(d)</td>
<td>diameter of the liquid jet</td>
<td>m</td>
</tr>
<tr>
<td>(D)</td>
<td>diameter of a droplet</td>
<td>m</td>
</tr>
<tr>
<td>(G_\varphi)</td>
<td>axial flux of angular momentum</td>
<td>Nm</td>
</tr>
<tr>
<td>(G_z)</td>
<td>axial flux of linear momentum</td>
<td>N</td>
</tr>
<tr>
<td>(I)</td>
<td>constant</td>
<td>(\sqrt{m})</td>
</tr>
<tr>
<td>(ISMD)</td>
<td>Integral Sauter Mean Diameter</td>
<td>(\mu m)</td>
</tr>
<tr>
<td>(L)</td>
<td>latent heat of vaporization</td>
<td>J/kg</td>
</tr>
<tr>
<td>(LHV)</td>
<td>lower heating value</td>
<td>MJ/kg</td>
</tr>
<tr>
<td>(\dot{m})</td>
<td>mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>(M)</td>
<td>molecular weight</td>
<td>kg/kmol</td>
</tr>
<tr>
<td>(MKE)</td>
<td>mean kinetic energy</td>
<td>J/kg</td>
</tr>
<tr>
<td>(p_g)</td>
<td>pressure</td>
<td>bar</td>
</tr>
<tr>
<td>(r)</td>
<td>radial coordinate</td>
<td>m</td>
</tr>
<tr>
<td>(R)</td>
<td>radius of the mixing tube</td>
<td>m</td>
</tr>
<tr>
<td>(R^2)</td>
<td>coefficient of determination</td>
<td>-</td>
</tr>
<tr>
<td>(SMD)</td>
<td>Sauter mean diameter</td>
<td>(\mu m)</td>
</tr>
<tr>
<td>(t)</td>
<td>time</td>
<td>s</td>
</tr>
<tr>
<td>(T)</td>
<td>temperature</td>
<td>K</td>
</tr>
<tr>
<td>(TKE)</td>
<td>turbulent kinetic energy</td>
<td>J/kg</td>
</tr>
<tr>
<td>(w)</td>
<td>velocity</td>
<td>m/s</td>
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<tr>
<td>(Y)</td>
<td>mass fraction</td>
<td>-</td>
</tr>
<tr>
<td>(z)</td>
<td>axial coordinate</td>
<td>m</td>
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Greek letters

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<th>Notation</th>
<th>Description</th>
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<tbody>
<tr>
<td>δ</td>
<td>mass diffusivity</td>
<td>m²/s</td>
</tr>
<tr>
<td>η</td>
<td>energy transfer efficiency</td>
<td>-</td>
</tr>
<tr>
<td>Λ</td>
<td>evaporation constant</td>
<td>m²/s</td>
</tr>
<tr>
<td>µ</td>
<td>dynamic viscosity</td>
<td>Pa s</td>
</tr>
<tr>
<td>ν</td>
<td>kinematic viscosity</td>
<td>m²/s</td>
</tr>
<tr>
<td>ρ</td>
<td>density</td>
<td>kg/m³</td>
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<tr>
<td>σ</td>
<td>surface tension</td>
<td>N/m</td>
</tr>
<tr>
<td>θ</td>
<td>thermal diffusivity</td>
<td>m²/s</td>
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Dimensionless numbers

<table>
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<tr>
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<th>Description</th>
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<tr>
<td>$ALR = \frac{m_A}{m_L}$</td>
<td>air-to-liquid mass flow ratio</td>
</tr>
<tr>
<td>$B_M = \frac{Y_{L,s}}{(1 - Y_{L,s})}$</td>
<td>mass transfer number</td>
</tr>
<tr>
<td>$B_T = \frac{c_{p,A}}{(T_\infty - T_s)}/L$</td>
<td>thermal transfer number</td>
</tr>
<tr>
<td>$Ev = \frac{t_{res}}{t_{evap}}$</td>
<td>Evaporation number</td>
</tr>
<tr>
<td>$Le = \frac{\theta}{\delta}$</td>
<td>Lewis number</td>
</tr>
<tr>
<td>$Ma = \frac{w_A}{c}$</td>
<td>Mach number</td>
</tr>
<tr>
<td>$MFR = \frac{\rho_A w_A^2}{\rho_L w_L^2}$</td>
<td>momentum flux ratio</td>
</tr>
<tr>
<td>$Oh = \frac{We^{1/2}}{Re} \equiv \frac{\mu}{\sqrt{\sigma \rho d}}$</td>
<td>Ohnesorge number</td>
</tr>
<tr>
<td>$Re = \frac{wd\rho}{\mu}$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$S = \frac{G_z}{G_2 R}$</td>
<td>swirl number</td>
</tr>
<tr>
<td>$Stk = \frac{(\rho_L - \rho_A)D(w_A - w_L)}{(18\mu_L)}$</td>
<td>Stokes number</td>
</tr>
<tr>
<td>$We = \frac{pu^2d}{\sigma}$</td>
<td>Weber number</td>
</tr>
<tr>
<td>$\lambda = \frac{\dot{m}<em>A}{\dot{m}</em>{A,sto}}$</td>
<td>air-to-fuel equivalence ratio</td>
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Chemical species

<table>
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<tr>
<td>$C_2^*$</td>
<td>excited C₂ radical</td>
</tr>
<tr>
<td>CH*</td>
<td>excited CH radical</td>
</tr>
<tr>
<td>$CO_2^*$</td>
<td>excited carbon dioxide</td>
</tr>
<tr>
<td>NOX</td>
<td>nitrogen oxides (NO and NO₂)</td>
</tr>
<tr>
<td>SOX</td>
<td>sulfur oxides (SO₂ and SO₃)</td>
</tr>
<tr>
<td>OH*</td>
<td>excited OH radical</td>
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Note that only those chemical species are listed here which differ from the conventional notation.
### Subscripts

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<tr>
<td>0</td>
<td>initial value</td>
</tr>
<tr>
<td>516</td>
<td>at 516 nm wavelength</td>
</tr>
<tr>
<td>554</td>
<td>at 554 nm wavelength</td>
</tr>
<tr>
<td>A</td>
<td>air</td>
</tr>
<tr>
<td>c</td>
<td>critical</td>
</tr>
<tr>
<td>e</td>
<td>end of the period</td>
</tr>
<tr>
<td>evap</td>
<td>evaporation</td>
</tr>
<tr>
<td>g</td>
<td>(atomizing) gauge (pressure)</td>
</tr>
<tr>
<td>hu</td>
<td>heat-up (period)</td>
</tr>
<tr>
<td>L</td>
<td>liquid</td>
</tr>
<tr>
<td>max</td>
<td>maximum value</td>
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<td>r</td>
<td>radial component</td>
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<td>R</td>
<td>relative</td>
</tr>
<tr>
<td>res</td>
<td>residence</td>
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<tr>
<td>s</td>
<td>at the drop surface</td>
</tr>
<tr>
<td>st</td>
<td>steady-state (evaporation period)</td>
</tr>
<tr>
<td>sta</td>
<td>static (pressure)</td>
</tr>
<tr>
<td>sto</td>
<td>stoichiometric</td>
</tr>
<tr>
<td>t</td>
<td>total (pressure)</td>
</tr>
<tr>
<td>z</td>
<td>axial component</td>
</tr>
<tr>
<td>ϕ</td>
<td>tangential component</td>
</tr>
<tr>
<td>∞</td>
<td>at infinity</td>
</tr>
</tbody>
</table>

### Abbreviations

<table>
<thead>
<tr>
<th>Notation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFD</td>
<td>computational fluid dynamics</td>
</tr>
<tr>
<td>IRZ</td>
<td>inner recirculation zone</td>
</tr>
<tr>
<td>DLE</td>
<td>dry low emission</td>
</tr>
<tr>
<td>FES</td>
<td>flame emission spectroscopy</td>
</tr>
<tr>
<td>LPP</td>
<td>lean premixed prevaporized (burner)</td>
</tr>
<tr>
<td>nMOS</td>
<td>n-channel metal-oxide semiconductor field-effect transistor</td>
</tr>
<tr>
<td>NT</td>
<td>Nukiya-Tanasawa (distribution function)</td>
</tr>
<tr>
<td>ORZ</td>
<td>outer recirculation zone</td>
</tr>
<tr>
<td>PDA</td>
<td>Phase Doppler Anemometer</td>
</tr>
<tr>
<td>PDF</td>
<td>probability density function</td>
</tr>
<tr>
<td>RME</td>
<td>rapeseed methyl ester</td>
</tr>
<tr>
<td>RR</td>
<td>Rosin-Rammler (distribution function)</td>
</tr>
<tr>
<td>UV</td>
<td>ultra violet</td>
</tr>
<tr>
<td>VIS</td>
<td>visible</td>
</tr>
<tr>
<td>Γ</td>
<td>Gamma (distribution function)</td>
</tr>
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</table>
Chapter 1

Objective and outline

The continuously growing fossil fuel utilization and the caused climate change made it clear that significant efforts have to be made to reduce the growth of carbon dioxide emission and achieve a sustainable economy through carbon neutrality. A few decades ago the energy sector envisioned that agriculture might help the situation and renewable fuels produced from crops can replace fossil fuels. However, it turned out shortly that cultivating even the whole surface of the Earth can cover only a portion of our energy demand. The renewable fuel technologies fall to the well-established fossil fuel based ones in a clean competition, but a political support might change the situation while they have a significantly lower ecological footprint. Therefore, the improvement of renewable combustion technologies has considerable potential and is highly relevant today.

The transportation sector (i.e., not exclusively the road, but air and marine applications as well) heavily relies on liquid fuels due to easy handling and operation safety. While there is no viable alternative solution, especially in air transportation, diluting the fossil fuel with renewable ones is the only possible scenario today towards carbon neutrality. Rapeseed produced in the largest volume in the European Union as fuel crop while various plants are yielded in other parts of the world for fuel production, e.g., soy, palm, and jatropha. The extracted vegetable oils are not used in its crude form but transesterified to biodiesel. A goal of the present thesis is to investigate crude vegetable oil, especially, crude rapeseed oil combustion and determine under which conditions it is equivalent to diesel oil combustion. By skipping the transesterification process, rapeseed oil utilization is more competitive to fossil fuels. The results can be used in boilers, furnaces, and thermal power plants besides transportation applications. Nevertheless, crude vegetable oil combustion requires steady conditions, therefore, they cannot be utilized in reciprocating engines.

Regardless of the origin of the fuel, continuously stringent decrees and standards maximize the allowed pollutant emission. Ideally, hydrocarbon flames emit only water and carbon dioxide. However, incomplete combustion results in unburnt hydrocarbons, carbon monoxide, and soot emission. NO$_X$ is another harmful pollutant formed by endothermic reactions at high temperatures and its concentration in the exhaust gas is also strictly limited. In order to be in line with the emission standards, lean combustion is widely applied today. It means that excess air is present in the combustion chamber beside the required amount for stoichiometric conditions. Therefore, the combustion will be complete as there is enough oxygen to burn the fuel. The remaining air reduces the overall combustion temperature due to its heat capacity, hence, reduces the NO$_X$ emission. The key features of liquid fuel combustion are summarized in Fig. 1.1.
Phenomena with thick border are investigated in the present thesis. The process starts with atomization which ensures low enough droplet sizes. These evaporate and mix with the combustion air before reaching the flame front. The latter condition is crucial to eliminate the hot spots which can lead to progressively increasing NO\textsubscript{X} emission. The combustion itself starts with the ignition of the fuel which is largely determined by the flow field and the fuel properties. If the burning velocity is higher than the flow velocity, the flame flashes back and may damage the burner itself. If the flow velocity is higher, the flame blows out. If the combustion chamber is not designed properly, acoustic-originated oscillations may damage the device. The term emission is not restricted to the pollutants. It also includes the radiative heat transfer to the combustion chamber walls, the chemiluminescence emission, and the combustion noise. Figure 1.2 demonstrates the flow pattern of a liquid fueled burner and a real combustion chamber.

Figure 1.1: Primary mechanisms and their relation of premixed combustion. Thick borders indicate the investigated phenomena.

Figure 1.2: Flow pattern of a preswirled burner (left, [1]) and section of a gas turbine combustion chamber (right, [2]).
Note that the burner is a preswirled type, in which the role of the swirl is to enhance mixing between the fuel vapor and the combustion air beside stabilizing the flame through the appearing vortex core. A section of an annular combustion chamber of a General Electric CF6-50 gas turbine is shown on the right. The liner is annular-shaped, and several fuel injectors are installed circumferentially. The flame is located inside the liner.

This thesis is divided into six chapters from which the second one reviews the state-of-the-art of the corresponding technical literature. Then the forthcoming ones detail the investigations, following the life of the fuel from the atomizer to the combustion chamber outlet. The paragraphs below detail them.

The combustion starts with atomization, discussed in Chapter 3, where the fuel breaks up into droplets with a typical diameter of several \( \mu m \). The process can be properly described neither analytically nor numerically. Therefore, the literature contains various empirical and semi-empirical formulas to estimate the volume-to-surface diameter or Sauter Mean Diameter, \( SMD \), while this is the most relevant from evaporation point of view. Here comes the first goal of the thesis: comparing various \( SMD \) estimating formulas with the measurement results of a plain-jet airblast atomizer. In this configuration, the fuel is flowing in a central pipe, and the high-velocity atomizing air is discharged from an annular nozzle concentric with that. The velocity difference leads to the breakup of the fuel jet into droplets. Regarding the size distribution, two sprays might be different even though having the same \( SMD \). Therefore, three different probability density functions compared while they are often used in the literature of atomization. The purpose of the investigation is similar to the previous case: finding out which one describes the spray more accurately.

It was mentioned above that the droplets have to evaporate before reaching the flame front. Chapter 4 details the evaporation characteristics of various fossil and renewable liquid fuels, using the so-called \( D^2 \) law. This name comes from the fact that the evaporation time is proportional to the square of the droplet diameter. The investigated burner has a subsequent tube after the atomizer which role is ensuring the evaporation of the fuel droplets and mix the vapor with the combustion air. This tube is often referred to as mixing tube. Based on the atomizing conditions, it is possible to calculate the residence time of the droplet inside the mixing tube. Assuming that the flame front is located after it and knowing the average droplet size of atomization, the ratio of the droplet residence time to the evaporation time can be calculated. This quantity shows that the given droplet could evaporate within the mixing tube or not. It is crucial in the case of crude vegetable oils while they are less volatile than the conventional fossil fuels. Consequently, their characteristics are compared to that of standard diesel oil. By preheating the vegetable oils, their surface tension and viscosity can be lowered which improves their vaporization.

As it was mentioned above, to minimize the pollutant emissions, operating highly lean is attractive (assuming that the atomization and evaporation-related problems are overcome based on the previous two chapters). Nevertheless, combustion stability issues arise along with lean operation while using much diluting air leads to flame blowout as the flow velocity overcomes the burning velocity, shown in Fig. 1.3 by an SR-71 Blackbird reconnaissance aircraft. Chapter 5 discusses and compares the blowout stability of diesel and rapeseed oil combustion with special emphasis on the swirl characteristics. If there is no side flow, as it was shown in Fig. 1.2, an outer recirculation zone is also formed around the inner one. This can be ‘filled’ by solid material, resulting in a subsequent diffuser geometry. However, the appearing boundary layer on the wall of the diffuser might affect the blowout stability of the burner, which is discussed in detail. Regarding
the pollutant emissions, carbon monoxide and NOX of diesel and rapeseed oil combustion were investigated and compared to the actual Hungarian decree. The goal was here to determine under which conditions the rapeseed oil can substitute diesel oil.

To date, variable loading gains increasing attention in numerous combustion applications including energy production. Besides the competitive fuel consumption, fulfilling emission limitations is obligatory. Therefore, appropriate combustion control becomes highly relevant. The spectrometer is a potential device which can withstand the harsh industrial environment and provide real time data through an optical access. The investigated spectra typically vary from 300 nm up to 1100 nm. In this range, the most significant chemiluminescence emitters are OH* at 310 nm, CH* at 431 nm, and C2 at 516 nm, where * notes the excited radicals. The ratio of the chemiluminescence emission peaks is proportional to the local air-to-fuel equivalence ratio (i.e., how much times more air is present at the location than it is required for the stoichiometric reaction). This methodology is carried out for both diesel and rapeseed oil combustion to investigate the change of radical ratios, discussed in Chapter 6.

The thesis contains four appendices which detail the measurement uncertainty of the used devices and their data sheet, the calculation of the swirl number, the temperature-dependent material properties used for calculating the evaporation, and high-speed sequences of the flame blowout mechanism with and without a diffuser.
Chapter 2

Introduction

2.1 Combustion of liquid fuels

Efficient combustion of liquid fuels in heat engines dates back to the end of the 19th century. A new era has begun by the invention of carburetor by Donát Bánki and János Csonka. In the 20th century, the most significant milestone was the development of highly efficient combustion systems [4]. They were characterized by less unburnt fuel and particulate matter emission. Today, the practical issues of heat engines are mostly solved. Consequently, the efficiency of fossil fuel combustion reaches almost 100% [2] (i.e., the conversion of chemical energy to thermal energy). The crude oil supplies are finite which lead to the challenge of the 21st century, namely, achieving a sustainable economy. Currently, there are emerging technologies, which aim to utilize crops as fuel in a large extent. Therefore, food safety requires determining the best ratio of crop usage between food and biofuel [5]. As long as electric, hybrid, and fuel cell systems struggle with technological issues and low primary fuel-to-wheel efficiency, combustible liquid fuels will dominate mainly the transportation sector [6]. Power generation also benefits considerably from such studies as it improves the supply safety [7]. However, governmental support significantly affects the used technologies.

Regardless of the most reasonable step being the increase of the renewable content of the standardized fuels, the present PhD thesis compares the utilization of crude vegetable oils with standard diesel oil (according to EN 590:2014) in a steady operating burner. The reason behind that is mixing always results in more favorable combustion characteristics [8,9]. Consequently, the worst-case scenario is being investigated here. By transesterifying crude vegetable oils, Fatty Acid Methyl Esters (FAMEs) are produced which is the biodiesel. Their physical properties, e.g., viscosity, surface tension, and density are very close to those of diesel oil. However, they have less heating value due the natural presence of oxygen at the end of the hydrocarbon chain. Apparently, by skipping the transesterification process, the production cost of the fuel decreases significantly. To date, very few studies are available regarding the utilization of crude vegetable oils in heat engine environment (see, e.g., [7,8,10]), unlike the investigation of FAMEs. The reason is trivial: decrees and standards force the oil companies to add FAME to the diesel oil all around the globe. In the European Union, the EN 590:2014 diesel oil standard allows maximum 7% FAME content by volume. It is fulfilled primarily by adding rapeseed methyl ester (RME). Globally, many other vegetable oils are used as additives after transesterification (e.g., soybean, palm, jatropha, and sunflower oil). The potential applications for crude vegetable oil utilization consist of gas turbines, furnaces, and boil-
ers. Due to the significantly less volatility of such fuels, it is cumbersome to utilize them directly in internal combustion engines [11].

2.2 Atomization and spray characterization

Liquid fuels have to be evaporated before combustion because they are flammable only in vapor form. Practically, they react with the oxygen content of the air. Therefore, atomizers are used in order to increase the surface of the fuel through shattering it into small droplets, facilitating evaporation. The atomization process is often accomplished by discharging high-velocity liquid to a relatively still air (pressure atomizers) or discharging high-velocity air next to the slow-velocity fuel (twin-fluid atomizers) [12]. The arising velocity difference leads to the breakup of liquid into droplets. Other atomizer types also exist, such as effervescent, electrostatic, ultrasonic, and whistle [13]. However, the present thesis investigates a burner equipped with a plain-jet airblast atomizer, which is a member of the twin-fluid family.

Air blast atomization was systematically examined by Nukiyama and Tanasawa first in 1939 [14], using a plain-jet airblast atomizer. The prefilming airblast atomizer was developed by Lefebvre and Miller in 1966 [15]. Both atomizers are shown in Fig. 2.1.

![Figure 2.1: Nukiyama and Tanasawa (left) and Lefebvre and Miller atomizer (right) [12].](image)

The importance of airblast atomizers is that they are capable of operating both at full load and idle conditions (i.e., the turn-down ratio is about 50:1, which is out of the capabilities of a pressure atomizer). Consequently, it is a favorable solution for gas turbines and other steady operating combustion devices. Besides heat engines, these atomizer types are widely used in other fields, such as metallurgy [16], painting, and coating technologies [17].

Measurement of spray characteristics started from magnesium oxide coated plates, liquid nitrogen freezing, atomization of molten wax, and high-speed photography [13,18]. Nowadays, Phase Doppler Anemometry (PDA) is used extensively for this purpose [19,20], discussed in Section 3.1. Also, emerging methods can be found in the literature, such as infrared thermography [21]. Most of the spray measurements available in the literature are carried out under atmospheric conditions. The significance of those is that the burner performance, including atomization and evaporation, will be more than adequate under elevated operating pressures. Therefore, the atomizers should be designed for the minimum pressure conditions [22]. A more recent measurement series of a full-scale gas turbine
atomizer from 1 to 12 bar pressure concluded that the operating pressure has negligible influence on spray quality [23] if the proper dimensionless quantities match. These are the Reynolds number, $Re$, the Weber number, $We$, the Ohnesorge number, $Oh$, and the momentum flux ratio, $MFR$, as follows:

$$Re = \frac{wd\rho}{\mu} = \frac{wd}{\nu}, \quad (2.1)$$

$$We = \frac{\rho w^2 d}{\sigma}, \quad (2.2)$$

In order to eliminate the dependence on $w$, the Ohnesorge number (or Z number) can be derived from $Re$ and $We$:

$$Oh = \frac{We^{1/2}}{Re} = \frac{\mu}{\sqrt{\sigma \rho d}} \quad (2.3)$$

$$MFR = \frac{\rho_A w_A^2}{\rho_L w_L^2}, \quad (2.4)$$

where $w$ is the velocity, $d$ is a typical dimension (e.g., the diameter of the fuel jet or a droplet diameter), $\rho$ is the density, $\mu$ is the dynamic, $\nu = \mu/\rho$ is the kinematic viscosity, and $\sigma$ is the surface tension. $A$ and $L$ subscripts note air and liquid, respectively. In Eqs. 2.1, 2.2, and 2.3 the viscosity and the density have no subscripts while they can be determined for both air and liquid phases. The velocity always means the relative velocity (i.e., air minus liquid). These dimensionless numbers have an $A$ or $L$ subscript, noting the basis of calculation clearly.

Based on the background phenomena, four primary liquid disintegration mechanisms are distinguished by the literature, shown later in Fig. 2.2. The various regimes also differ in the generated droplet sizes. According to Reitz [24], these are the following:

1. Rayleigh jet breakup is caused by the growth of axisymmetric oscillations of the jet surface, induced by the surface tension. Typically, the droplet diameter exceeds the jet diameter [25].

2. First wind-induced breakup. The surface tension is now augmented by the relative velocity between the jet and the ambient gas, thereby accelerating the breakup process. As in regime 1, breakup occurs many jet diameters downstream of the nozzle. Drop diameters are about the same as the jet diameter [26]. Nevertheless, small-size droplets, called satellite droplets, may form in both regime 1 and 2 through the disintegration process from the elongated liquid between two forming droplets. This is a rather challenging phenomenon to model.

3. Second wind-induced breakup. Drops are produced by the unstable growth of short-wavelength surface waves caused by the relative motion of the liquid jet and the ambient gas. The growth of these waves are opposed by the surface tension. Breakup occurs several diameters downstream of the nozzle outlet. The resulting droplet sizes are comparable with the most unstable wavelength, which is significantly smaller than the jet diameter [27].

4. Atomization. The jet disrupts completely at the nozzle outlet. Average droplet diameters are much less than the jet diameter. It occurs typically in the high-pressure atomizers of compression ignition engines. The most probable explanation of this phenomenon is the appearing cavitation zone inside the nozzle orifice [28].
These four regimes are shown in Fig. 2.2. It should be noted that the characteristic lines move towards the origin by increasing the operational pressure [24]. This effect has a higher importance in the automotive industry at chamber pressures of several tens of bars [29] and less relevant under typical gas turbine conditions.

Figure 2.2: Classification of liquid jet disintegration modes [13].

In the upcoming part of this section, liquid fuel spray characterization is discussed. Atomization of crude vegetable oils is cumbersome compared to that of conventional fossil fuels as they have higher kinematic viscosities and surface tensions due to their larger molecular sizes. These two physical properties are the major obstacles of adequate atomization, thus efficient combustion at room temperature. Therefore, preheating is necessary [30], and it is the most efficient means of preparing such fuels, discussed in Chapter 4.

This section is divided further into three subsections. While the literature consists of several empirical formulas for the estimation of different mean diameters, Subsection 2.2.1 highlights the most relevant ones. Often the smallest and the largest droplets have the most impact on the combustion performance and pollutant emissions. Therefore, it might be crucial to estimate the size distribution function, discussed in Subsection 2.2.2. Finally, a short summary of the droplet stability criteria is presented in Subsection 2.2.3 due to its importance in the evaluation of atomization measurement.

2.2.1 Estimation of the mean droplet diameter

In order to characterize a spray, different mean diameters are used in the literature, depending on the application. Most of them are based on the following formula:

\[ D_{jm} = \left[ \frac{\sum_{i=1}^{n} D^i_j}{\sum_{i=1}^{m} D^m_i} \right]^{1/(j-m)} \]  

where \( n \) denotes the number of measured droplets, \( j \), and \( m \) are typically integers. E.g., \( D_{10} \) is the arithmetical mean diameter. The Sauter Mean Diameter (SMD, \( D_{32} \), or volume-to-surface diameter) used mostly in the combustion literature due to its importance in evaporation. Droplet sizes cannot be derived neither analytically nor numerically for any practical atomizers even with the state-of-the-art tools. Therefore, semi-empirical
formulas are established for different atomizer configurations. The first goal of the thesis is to compare different SMD estimating formulas and extend their range of usability to high-speed atomization. It is detailed in Chapter 3 by measuring the generated spray of a plain-jet airblast atomizer. The fitted trends using the below-detailed equations are shown later in Fig. 3.11 of Chapter 3. All variables here refer to their values at the nozzle discharge. Lefebvre derived the following formula by parameter analysis and using vast amount of measurement data for SMD estimation of airblast atomizers [12]:

\[
\frac{SMD}{d_0} = (A W e^{-0.5} + B O h_L) \left( 1 + \frac{1}{A L R} \right),
\]

where \(d_0\) is the initial liquid jet diameter, \(W e_A\) is the Weber number based on the properties of air, \(O h_L\) is the Ohnesorge number based on the properties of liquid, and \(A L R\) is the air-to-liquid mass flow ratio. \(A\) and \(B\) are constants. Note that the velocity in \(W e_A\) is the relative one, \(w_R\), instead of \(w_A\). However, the measurements of Rizk and Lefebvre [31] showed better correlation with a modified version of Eq. 2.6:

\[
\frac{SMD}{d_0} = C \left( \frac{\sigma}{\rho_A w^2 d_0} \right)^{0.4} \left( 1 + \frac{1}{A L R} \right)^{0.4} + E \left( \frac{\mu^2}{\sigma \rho_L d_0} \right)^{0.5} \left( 1 + \frac{1}{A L R} \right),
\]

Equation 2.7 is a widely recognized formula in the literature, using the following values for the constants: \(C = 0.48\) and \(E = 0.15\) [2, 7, 32]. Subscripts \(A, L, R,\) and \(0\) refer to air, liquid, relative, and initial value (i.e., at the nozzle discharge), respectively. \(\sigma\) is the surface tension, \(\rho\) is the density, \(w\) is the velocity, \(d\) is the diameter of the liquid jet, and \(\mu\) is the dynamic viscosity. Eq. 2.7 was developed for kerosene, diesel oil, and blend oils at \(w_A = 10 - 120\) m/s and \(p_0 = 0.01 - 0.077\) bar. Contrary to that, Eq. 2.7 was used in the literature for the same burner which is the subject of the present thesis at conditions that are beyond the range of validity [7, 32–34]. Another formula for airblast atomization was derived by Lorenzetto and Lefebvre [35]:

\[
\frac{SMD}{d_0} = \left[ F \frac{(\sigma m_L)^{0.33}}{w_R \rho^0.33 \rho_L^{0.30} d_0} + H \left( \frac{\mu^2}{\sigma \rho_L d_0} \right)^{0.5} \right] \left( 1 + \frac{1}{A L R} \right)^{1.7},
\]

where \(F\) and \(H\) are constants and \(\dot{m}\) is the mass flow rate. Their finding for the constants were: \(F = 0.95\) and \(H = 0.13\). Albeit this formula was derived partially by Lefebvre, he later on referred to Eqs. 2.6 and 2.7 that they are more accurate as they rely on a more extensive measurement data. Note that the second term in Eq. 2.8 is \(O h_L\), as it is present in Eqs. 2.6 and 2.7. Varga et al. published a more recent formula for SMD estimation [36], based on the growth of the most unstable waves (second wind-induced atomization regime at the discharge):

\[
\frac{SMD}{d_0} = \left[ d_0 \rho_A^{0.75} \sqrt{w_A \left( 1 + \sqrt{\rho_A/\rho_L} \right) - w_L} \right]^{0.25},
\]

where \(I\) is a constant with a dimension of \(\sqrt{w}\) and \(\nu\) is the kinematic viscosity. They found that \(I = 0.055\) \(\sqrt{w}\) gave the best fit. \(I\) was not connected to any physical quantity, however, it seems to be in relation with the initial jet diameter. By comparing Eq. 2.9 with Eqs. 2.6–2.8, the most significant difference is the exclusion of the liquid viscosity. If the atomizing air discharge velocity is high enough, the viscosity has no time to affect the process. Lefebvre detailed this phenomenon in the case of prefilming airblast atomizers.
in 1992 [37]. He named the process prompt atomization. Regardless that there were no physical limits named, the background ideas contributed significantly to the understanding of high-velocity atomization. Moreover, he left a yet unsolved challenge for plain-jet airblast atomizers. Namely, checking the fit of Eq. 2.10:

\[ \frac{SMD}{d_0} = \frac{3}{2 + JW_{E_L}/[4(1 + 1/ALR)]}, \]  

(2.10)

where \( J \) is a constant practically, but it is in connection with the atomization efficiency. While the presently investigated pressure regime was \( p_g = 0.3 - 3.1 \) bar, it is expected that this value should be the function of at least the pressure of atomizing air or the \( MFR \). Here comes the formula of Park et al. [20] into the view:

\[ \frac{SMD}{d_0} = \frac{12}{8 + W_{E_L}/[1 + (1 + 1/\eta)]}, \]  

(2.11)

where \( \eta \) is the energy transfer efficiency that is now depending on the atomizing pressure as follows:

\[ \eta = K \bar{m}_L^{0.773} \left[ (p_t/p_{sta})^3 - 15.1 (p_t/p_{sta})^2 + 65 (p_t/p_{sta}) \right], \]  

(2.12)

where \( K \) is a constant, \( p_t \) is the total pressure, and \( p_{sta} \) is the static pressure. The viscosity term is also missing from Eqs. 2.11 and 2.12. The fitted polynomial may limit the application of Eqs. 2.11 and 2.12 to similar configurations to the investigated one. In their particular case, the atomized liquid was water at \( p_g = 1 - 4 \) bar. The nozzle geometry consisted of a 5 mm central pipe for water and an annular air discharge with 10 mm outer and 8.4 mm inner diameter. Table 2.1 summarizes the test conditions of the presented formulas. ND denotes 'not determined'.

<table>
<thead>
<tr>
<th>Equation</th>
<th>Author</th>
<th>Constants</th>
<th>( w_A ) [m/s]</th>
<th>( SMD ) [µm]</th>
<th>Liquid</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.6</td>
<td>Lefebvre [12]</td>
<td>ND</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2.7</td>
<td>Rizk &amp; Lefebvre [31]</td>
<td>0.48/0.15</td>
<td>10–120</td>
<td>15–110</td>
<td>kerosene, diesel oil, blended oils</td>
</tr>
<tr>
<td>2.8</td>
<td>Lorenzetto &amp; Lefebvre [35]</td>
<td>0.95/0.13</td>
<td>60–180</td>
<td>20–130</td>
<td>water, kerosene, special solutions</td>
</tr>
<tr>
<td>2.9</td>
<td>Varga et al. [36]</td>
<td>0.055 ( \sqrt{m} )</td>
<td>30–165</td>
<td>16–175</td>
<td>water, ethanol</td>
</tr>
<tr>
<td>2.10</td>
<td>Lefebvre [37]</td>
<td>ND</td>
<td>25–60</td>
<td>40–200</td>
<td>water, heating oil</td>
</tr>
<tr>
<td>2.11</td>
<td>Park et al. [20]</td>
<td>ND</td>
<td>328–469</td>
<td>6–52</td>
<td>water</td>
</tr>
</tbody>
</table>

Table 2.1: Test conditions of the discussed \( SMD \) estimating formulas.

It is known that the Design of Experiments mathematical concept may significantly reduce the number of required measurement setups if the guessed equation is properly determined and only constants are required. However, the derived equation by Chen et al. [38] is neglected here while they excluded the surface tension which is wrong. It is evident that it plays a major role in airblast atomization. Chen et al. relied on the work of Chin [39] for this statement who investigated a pressure-swirl atomizer that relies remarkably different discharge, hence, primary atomization mode.

Although there were several other formulas developed before the 1980s, the available measurement techniques considerably limited the detection of the small droplets at those times. Hence, the thesis confines to results which were based on laser measurement
techniques. The curious reader is redirected here to Lefebvre [12] for technical details and earlier measurement methods.

### 2.2.2 Droplet size distribution functions

Two sprays characterized by the same SMD may differ considerably. Moreover, it is the smallest and the largest droplets that are of paramount importance in many practical applications, which cannot be derived from the SMD by any means. The small droplets facilitate easier ignition by a short evaporation time while large droplets may reduce the combustion efficiency and increase pollutant emissions if they cannot completely evaporate before reaching the flame front. Therefore, droplet size distribution functions come into view [13]. This property received less scientific focus compared to the estimation of SMD [40]. The literature distinguishes the empirical method, the maximum entropy, and the discrete probability function methods for characterizing the droplet size distribution [40]. Even though the second and the third of the these methods rely on physical laws, none of them has so far been able to effectively compete with the empirical formulas [41–43]. Tharakan et al. [43] expects that Computational fluid dynamics (CFD) simulations will lead to a better understanding of the atomization process, but only in the future. As a consequence, the present thesis also aims to analyze empirically the fit of different probability density functions, PDFs, versus atomizing pressure. Typically, air-blast atomization is characterized by gamma archetype distribution functions [44]. Equations 2.13–2.15 present the three PDFs that are fitted to the measurement presented in Chapter 3. These are the gamma, (Γ), the Rosin-Rammler (RR, also known as Weibull), and the Nukiyama-Tanasawa (NT) PDFs, respectively.

\[
f(D)_\Gamma = \frac{D^{a-1}}{b^a \Gamma(a)} e^{-D/b} \tag{2.13}
\]

\[
f(D)_{RR} = \frac{b}{a} \left( \frac{D}{a} \right)^{b-1} e^{-(D/a)^b} \tag{2.14}
\]

\[
f(D)_{NT} = aD^g e^{-bD^g} \tag{2.15}
\]

Equations 2.13 and 2.14 are two-parameter PDFs \((a, b)\) while the Nukiyama-Tanasawa is a four-parameter one \((a, b, g, q)\). Here, \(g = 2\) was assumed in Eq. 2.15, according to previous results available in the literature [13,45]. Therefore, the NT PDF was investigated as a three-parameter one. In Section 3.5, these functions are fitted to the measurement results to derive the respective parameter values at various atomizing pressures. The results were summarized in Figs. 3.12–3.14.

### 2.2.3 Droplet stability

The initial disintegration of the liquid jet is called primary atomization. However, the resulting liquid fractions may undergo a secondary atomization if the critical Weber number is exceeded. Its calculation is based on Eq. 2.2 and its critical value is estimated to be 12 [2]. In the present case, it is suggested to consider both the effect of shear and the turbulence, noted with \(sh\) and \(tu\) subscripts, respectively [46]. Hence, these Weber numbers are [46,47]:
\[ We_{sh} = \rho_L (w_A - w_L)^2 D/\sigma, \quad (2.16) \]

\[ We_{tu} = 2\rho_A \varepsilon^{2/3} w^2 D^{5/3}/\sigma, \quad (2.17) \]

where \( \varepsilon \) is the turbulent dissipation rate and calculated as follows:

\[ \varepsilon = TKE^{3/2}/l, \quad (2.18) \]

where \( l \) is the turbulent length scale, estimated to be 3.8% of the nozzle hydraulic diameter in the case of annular flows [48]. \( TKE \) is the turbulent kinetic energy, determined from the PDA measurement data:

\[ TKE = 0.5 \left[ \overline{(w_z')^2} + \overline{(w_r')^2} + \overline{(w_\phi')^2} \right], \quad (2.19) \]

where \( z, r, \) and \( \phi \) subscripts representing the axial, radial, and tangential components of the flow velocity, respectively. Primes are the fluctuations around the temporal average while overbars indicate the ensemble average. Now the maximum size of the stable droplet can be determined:

\[ D_{max} = \min \left\{ \frac{\sigma We_{sh,c}}{\rho_L (w_A - w_L)^2}, \left[ \frac{\sigma We_{tu,c}}{\rho_A} \right]^{3/5} \varepsilon^{-2/5} \right\}, \quad (2.20) \]

where subscript \( c \) denotes the critical values. \( We_{sh,c} = 12 \) and \( We_{tu,c} = 0.59 \) were used in the present paper based on literature data [13, 49]. According to the work of Hinze [49], breakup of a single diesel oil droplet was investigated in high-speed gas stream under atmospheric conditions. Therefore, the mentioned critical \( We \) number values can be applied in a general case, regardless of the atomizer configuration.

### 2.3 Evaporation of droplets

Obviously, atomization is crucial in liquid fuel combustion, but its aim is only to enhance the vaporization rate to ensure the fuel vapor-air mixture before reaching the flame front. Nevertheless, infinitely small droplets do not necessarily improve combustion efficiency. If the evaporation is complete and a homogeneous fuel-air mixture reaches the flame front, it does not count anymore how small was the average droplet size and how fast could the spray evaporate. The present section details the theoretical background of the evaporation phenomena.

Droplet evaporation includes simultaneous heat and mass transfer, where the heat is transferred from the surrounding gas to the drop surface by convection and conduction. The vapor is transferred back to the gas stream by convection and diffusion. The key parameters of evaporation are the pressure, temperature, and transport properties of the gas [13]. If the fuel droplet is not fully evaporated when its vapor ignites, diffusion flame can be observed around it. If the evaporation is complete, furthermore, the vapor is mixed well with the air, then a premixed flame is formed. Obviously, the latter one is preferred in order to have high combustion efficiency and low pollutant emissions, as discussed in Section 2.4.

Historically, the increasing interest in rocket and gas turbine technologies led to rapid development in the field of evaporation characterization. The first two comprehensive
studies are related to Godsave [50] and Spalding [51] in 1953. The method is simply referred to as $D^2$ law in the combustion literature, shown later in Eqs. 2.21–2.28. In order to simplify the evaporation calculation, the following assumptions are made:

- The droplet is spherical.
- The fuel is a pure liquid having a well-defined boiling point.
- Radiation heat transfer is negligible.
- $Le = \theta/\delta = 1$,

where $Le$ is the Lewis number, the ratio of thermal ($\theta$) and mass diffusivity ($\delta$). These conditions are claimed to be in line with practice, except the case of highly luminous or very low pressure flames [13].

The lifetime of a droplet can be divided into two stages: the heat-up period, which is followed by the steady-state evaporation, shown in Fig. 2.3. A portion of the incoming heat also result in evaporation during the heat-up period besides heating the liquid phase itself. Nevertheless, its extent is mostly related to the volatility of the liquid, the ambient, and the initial conditions [13]. The duration of the two phases is the total time of evaporation.

Figure 2.3: Theoretical model for the evaporation of a single droplet [13].

Since the work of Godsave and Spalding, many researchers contributed to evaporation theory, see, e.g., [52–57]. In the 5th International Workshop on Measurements and Computation of Turbulent Spray Combustion, Rhodes, Greece, 2015 famous researchers joined to discuss key issues of liquid combustion which was the target of the previous session in 2013. The conclusion was clear regarding evaporation. There were numerous models developed in the past decades with the help of advanced laser measurements and CFD. All of them underestimated the evaporation time compared to a wide range of measurements. There was only one model, which overpredicted that, namely, the $D^2$ law. The same conclusion was made by Driscoll [58], who presented the development experiences of the Rolls-Royce Trent 1000 gas turbine in the 9th Mediterranean Combustion Symposium, Rhodes, Greece, 2015. Consequently, the present PhD thesis is confined to the use of $D^2$ law to remain conservative. The key equations are listed below and are an excerpt from the Atomization and Sprays textbook [13].

\[ D_0^2 - D_{hu,x}^2 = \Lambda_{hu} t_{hu}, \]  

\[ \text{Figure 2.3: Theoretical model for the evaporation of a single droplet [13].} \]
where \( D \) is the initial droplet diameter and subscripts 0, \( hu \), and \( e \) refer to initial, heat-up, and end of the period, respectively. \( \Lambda \) is the evaporation constant and \( t \) is the time. Equation 2.21 describes the heat up phase, containing the reduction of the initial droplet diameter through evaporation. Here, \( D_0 \) is assumed to be equal to the SMD, therefore an average-sized droplet by volume-to-surface will be the subject of the present investigation. There are three unknowns in Eq. 2.21, therefore, two additional equations are required.

\[
\Lambda_{hu} = \frac{8k_A \ln (1 + B_M)}{c_{p,A} \rho_L}, \quad (2.22)
\]

\[
t_{hu} = \frac{c_{p,L} \rho_L c_{p,A} D_{hu,e}^2 (T_{s,ST} - T_{s,0})}{12k_g \ln (1 + B_M) L (B_T/B_M - 1)}, \quad (2.23)
\]

where \( c_p \) is the specific heat capacity at constant pressure, \( T \) is the temperature and subscripts \( s \) and \( st \) refer to the droplet surface and the steady-state evaporation period, respectively. \( L \) is the latent heat of vaporization, \( B_M \) is the mass, and \( B_T \) is the thermal transfer number, defined by Eqs. 2.25 and 2.26. \( k \) is the temperature-dependent thermal conductivity and subscript \( g \) denotes the gaseous mixture. The calculated heat conductivities were averaged by their mass fraction at a reference temperature, \( T_{ref} \), using Eq. 2.24.

\[
T_{ref} = T_{s,hu} + \frac{T_{\infty} - T_{s,hu}}{3}. \quad (2.24)
\]

\[
B_M = \frac{Y_{L,s}}{1 - Y_{L,s}} \quad (2.25)
\]

\[
B_T = \frac{c_{p,A} (T_{\infty} - T_s)}{L} \quad (2.26)
\]

Now Eq. 2.26 contains only known properties. Nevertheless, the mass fraction of the liquid at the surface, \( Y_{L,s} \), still has to be calculated. Eq. 2.27 puts an end to the set of equations:

\[
Y_{L,s} = \frac{p_{L,s}M_L}{p_{L,s}M_L + (p_{sta} - p_{L,s}) M_A}, \quad (2.27)
\]

where \( M \) is the molecular weight. Finally, the steady-state period has to be calculated until the droplet completely evaporates:

\[
D_{hu,e}^2 = \Lambda_{st} t_{st}. \quad (2.28)
\]

Equation 2.28 features \( \Lambda_{st} \), which can be calculated in the same way as \( \Lambda_{hu} \). While all the incoming heat results in the evaporation of the liquid under steady-state conditions, \( B_M = B_T \). Hence, \( B_T \) may be used instead of \( B_M \), if desired.

Based on the PDA measurements, detailed in Chapter 3, it was found that the densest spray region is located near the nozzle. Here, the estimated liquid-to-air ratio by volume is 34.21% which is confined to 10 mm downstream from the nozzle, near the axis. Its typical value was below 1% in the most cases. Consequently, it is assumed that there is no interaction between the individual droplets. Hence, evaporation of a single droplet fairly represents the evaporation of the whole spray.

Regarding the fuels, the evaporation characteristics of liquid fossil fuels are well-known [13]. As for rapeseed oil, Laza et al. [59] investigated the evaporation of the crude rapeseed
oil and some blends with alcohols. Their conclusion was that the $D^2$ law appropriately describes the evaporation process. Only a slight difference was recognized at the beginning of the heat-up period, namely, it starts with the expansion of the droplet due to the decreasing density with the increasing temperature. To incorporate their results, a new model would be required which is not among the goals of the present thesis but a possible future successor of this work.

2.4 Combustion in modern burners

Since the number of operating heat engines is continuously growing from one hand, and the climate change is proven to correlate with human activity on the other hand, combustion engineers have to find the best solutions to achieve high combustion efficiency. Furthermore, the continuously stricter emission standards put an additional severe pressure on the development, discussed in Subsection 2.4.3. However, there is no regulation on efficiency; customers prefer lower specific fuel consumption. Adequate performance and low pollutant emission at partial loading has become a critical issue nowadays. Consequently, combustion stability plays a more significant role than ever. The governing phenomena are summarized in Subsection 2.4.2. Designing appropriate flow pattern, discussed in Subsection 2.4.1, is not a direct constraint. Nevertheless, it is in strong connection with pollutant emissions and flame stability, therefore, it is the first subsection here. It introduces the background phenomena to understand stability and pollutant emission-related issues deeply.

There are far more papers available on atmospheric combustion experiments than on measurements at elevated pressure. The reason is that if the combustion performance is satisfactory at lower pressures, the situation will only improve at higher pressures [2,22]. This finding is consistent with the experiments of B. Higgins et al. [60,61]. Consequently, atmospheric tests which were carried out in the present thesis, might be directly applicable even for high-pressure combustion applications, such as gas turbines.

To date, fossil fuel utilization in heat engines is a routine work; the realized combustion efficiency is practically 100% [2]. Nevertheless, sustainable economy requires a change to renewable fuels [62]. Straight vegetable oils are characterized by low production cost compared to other liquid biofuels. Their use in reciprocating engines may lead to insufficient combustion performance due to clogging and delayed ignition (lower Cetane number) [11]. But steady-operating burners are capable of operating on them. Note that contaminations [7,63], malfunction [64] or even failure [65] may occur if the combustion chamber is not designed appropriately. Therefore, more data is required in order to understand the design requirements of vegetable oil combustion. There is wide agreement among the researchers on that fuel preheating is necessary [7,10,30,63,64]. Following atomization, the droplets travel towards the flame front, evaporate and mix with the combustion air in an ideal case. If not, the flame can become uneven, and the appearing hot spots decrease the maintenance intervals and increase the emission of nitrogen oxides (NOx) [66].

2.4.1 Flow patterns

The released chemical energy during combustion is several magnitudes higher than the kinetic energy of the flowing medium [65]. However, an appropriate flow field is essential to supply the flame with combustible mixture. Furthermore, mixing the reactants ensures low pollutant emission and even flue gas temperature profile which is necessary for gas
turbine and boiler technology to ensure high-efficiency combustion [66]. Consequently, all the upstream, the flame, and the downstream flow pattern have to be well-designed. A preferred solution for mixing the fuel and the air is the application of swirling flow. If the swirl is strong enough, the adverse pressure gradient along the jet axis cannot be further overcome by the kinetic energy of the fluid particles flowing in the axial direction. Hence, a toroidal-shaped recirculation zone is formed, characterized by two stagnation points [1]. It is called as internal recirculation zone (IRZ) or central recirculation zone. The diverging flow due to the IRZ forms an outer recirculation zone (ORZ) or external or corner recirculation zone. Both the IRZ and the ORZ are shown in Fig. 2.4.

Figure 2.4: IRZ and the ORZ at a) 10 kW and b) 35 kW combustion power, indicating the absolute velocity (left) and the tangential velocity (right) [67].

In Fig. 2.4 the combustion power has a little effect on the flow field and the flame length [1,68]. The role of the recirculation zones is to transfer heat from the flame to the upstream mixture, facilitating the evaporation of liquid fuels, ignition, decreasing the flow velocity, and preheating the mixture. Furthermore, the intense shear results in enhanced mixing [69].

Being a dimensionless quantity, the swirl number, \( S \), is widely used for describing swirling flows, hence swirling flames (see, e.g., [70–72]). It expresses the ratio of the axial flux of the angular momentum, \( G_\varphi \), and the axial thrust, \( G_z \), as follows [1]:

\[
S = \frac{G_\varphi}{G_z R}
\]  

(2.29)
where $R$ is the radius of the burner. This dimensionless number was first proposed by N. A. Chigier and J. M. Beér [73]. The calculation method of the swirl number can be found in Appendix B in the case of the investigated LPP burner. The swirl is considered strong if $S > 0.6$ [1]. At low swirl numbers, the flame is straight, while strong swirl results in a V-shaped flame through the appearing IRZ by the breakdown of the precessing vortex core [74], shown in Figs. 2.4 and 2.5. Note that the latter figure is an own result. If the OZR is strong enough, the flame front may shift further to M-shaped under certain conditions [75, 76] which may damage the burner [77].

![Figure 2.5: Straight (left) and V-shaped flame (right).](image)

![Figure 2.6: Comparison of the flow field with and without a diffuser [1].](image)

Applying diffusers on the burner can replace the ORZ in the case of V-shaped flames
while the flow field remains similar [78]. However, a remarkable difference is that a boundary layer will appear next to the wall of the diffuser. If the flow field is free from swirl, a larger IRZ is formed in this way [69], shown in Fig. 2.6 [1].

One can find the application of diffusers in the literature named as burner quarl, burner tile, conical outlet, and diverging nozzle. The first two terms are more specific to combustion technology among these names. The purpose of its use is to enhance flame stability [79]. The most common application of them is industrial furnaces for pulverized coal combustion, see, e.g., [80–84]. Nevertheless, the available literature is limited especially to solid fuel combustion, according to the best knowledge of the author. Among them, there is no comprehensive analysis regarding the optimal geometry. Despite this fact, it is claimed in a review paper that such studies are available [69] but supported very weakly in the reference list. Practically, most studies involve only one setup [80,84–87]. According to the best knowledge of the author, only El-Mahallawy et al. [88] carried out experiments using diffusers with various angles. They utilized liquefied petroleum gas without swirl and under rich conditions. The investigation of various half cone angles was not exhaustive, while they ended up with 30° that performed best. Moreover, the flame was not swirled. Consequently, application of diffusers with various angle is the subject of Subsection 5.3.2, utilizing liquid fuel under lean conditions and swirling flow. The application of bluff bodies for stabilization purposes attracted more scientific attention due to their use in, e.g., the afterburners of military gas turbines [1,3]. Hence, the forthcoming subsection discusses the stability of combustion.

2.4.2 Combustion stability

Providing stable flame front over a wide range of operating conditions is an obvious requirement of combustion chambers. Usually, it means various air-to-fuel ratios, flow rates, inlet pressures, and temperatures among other design and off-design situations. There are few different ways to realize unstable combustion. Neglecting the design related problems (i.e., resonance and inappropriate mixing [65]), lean and rich blowout characteristics limit the presence of a stable flame front as well as flashback and blowout. These lead to the fact that the term combustion stability is not well-defined in the literature; it can mean all the situations listed above [2]. For clarification, the subject of the present thesis is the investigation of blowout stability; all other aspects are currently neglected. The reason behind neglecting flashback is the design of the burner, shown if Fig. 4.1 in Chapter 4. The flame is not able to spread backward, and the application of a plain-jet airblast atomizer increases the operation safety even further. If the atomizing pressure is high and the combustion air flow rate is low, a large portion of the flame may be present inside the mixing tube. However, a boundary layer created by the combustion air protects the tube from melting down even in this situation. An additional reason is the lean operation near the blowout limit, which ensures low pollutant emission [65], discussed in Subsection 2.4.3. Note that flames near the blowout limit susceptible to lose stability at even few percent fluctuations in the equivalence ratio [89]. This is the price of operating lean [90]. Vortex shedding-related instability is known in the literature [90] which occurs at low flow velocities. While it was not observed during the numerous measurement series, it is also neglected in the present thesis.

Typically, the flame stabilization is enhanced by a diffuser or a bluff body [3,91–93]. Currently, only diffusers are investigated in Section 5.2. Nevertheless, flame stability can be improved by co-firing of a higher-grade fuel [94]. A highly attractive solution is
swirling the flow combined with all the methods above or used alone [1,69,94,95]. It acts similar to the application of a bluff body while the hot recirculated gas becomes the flame holder [95], shown in Fig. 2.5. When a diffuser is applied in addition to the swirl, the IRZ may enlarge [69]. Hence, it is expected that this combined solution affects the blowout stability limits favorably, discussed in Section 5.3. If the geometry should not be modified, the following two ways are followed to enhance the flame stability: applying either active control (e.g., loudspeakers, actuated parts) or passive elements (e.g. Helmholtz resonators, specially staged combustion chamber). Despite the wide variety of solutions, the industry prefers the passive ones to avoid additional failure possibilities [65].

It is known that the increased pressure widens the operating range of a burner. Therefore, the stable region at an elevated pressure will be at least the same as it is under ambient conditions [2,22], shown in Fig. 2.7.

![Figure 2.7: Influence of atomizing pressure on flame stability [2].](image)

However, ignition characteristics were not measured; it should be noted that it is possible in a limited regime compared to stable combustion. Therefore, relight near the stability limits might be cumbersome, shown in Fig. 2.8.

![Figure 2.8: Ignition loop of a burner [2].](image)

It is time to connect the stability-related problems to the flow patterns, discussed
earlier in Subsection 2.4.1. Regardless that the exact phenomena required to trigger flame blowout are not known [3], several observations may help to detect the key issues, detailed in Section 5.2. Premixed flames, which are investigated in this thesis, behave more flexible to unsteady phenomena. They have hot product gasses which remain close to the reactants and quickly reignite any quenched regions [96]. A swirl-stabilized flame above the critical swirl number is characterized by an IRZ and an ORZ, as it was discussed previously. The flame remains stable when the recirculated heat is in balance with the heat absorbed by the upstream mixture. If the blowout stability is near, holes and edges appear in the flame front typically near the burner lip due to the intensifying shear [3]. Hence, the balance between the recirculated and absorbed heats breaks. These phenomena lead to the cooling and shrinking of the IRZ, which is a self-excited process. Depending on the conditions, the flame blows out within few milliseconds.

2.4.3 Pollutant emissions

Pollutant emission is a critical, partially combustion-related issue that has a significant impact on both the environment and human health. Consequently, laws, decrees, and standards limit their maximum concentration in the exhaust gas, using a band system for different devices and thermal input powers. In order to minimize the pollutant emissions, there are two solutions: reduce their formation in the combustion chamber or use an after treatment, such as catalytic reduction. The present thesis is confined to its measurement and finding general trends during the operation of a burner, while the regulations may differ worldwide. E.g., the standards of California are among the strictest ones around the globe while the local crude oil contains \( \sim 0.49 \text{ m/m\% fuel-bonded nitrogen} \) facilitating the formation of NO\(_X\) [97]. Therefore, the allowed concentration is 9 ppm at 15% O\(_2\) in this state [66]. The actual Hungarian decree for gas turbines of >140 kW combustion power, 7/1999 (VII. 21.) KöM Decree, allows 200 mg/m\(^3\) NO\(_X\) and 100 mg/m\(^3\) CO emission at 15% O\(_2\). This latter limitation is used for evaluation in Section 5.4.

It is noteworthy that air-to-fuel equivalence ratio, \( \lambda = \dot{m}_A / \dot{m}_{A,sto} \), is used in this thesis instead of the more common fuel-to-air equivalence ratio. The former is more reasonable in combustion technology as the emission standards can be efficiently fulfilled under lean conditions [66].

Figure 2.9 presents the variation of the main pollutants versus the equivalence ratio [98]. At the first look, it becomes evident that excess air is required to keep the concentration of all pollutants low. However, understanding the formation of them is necessary to design the appropriate combustion chamber for a given application.

When the combustion zone is operating fuel rich, the flue gas will contain a large amount of CO, hydrocarbon radicals, and aromatic compounds instead of H\(_2\)O, CO\(_2\), and pure fuel. Reaching CO through the oxidation of hydrocarbons is a relatively fast process. Nevertheless, the oxidation of CO requires high temperature and OH or at least H\(_2\)O. Its main oxidation reactions are the following:

\[
\text{CO} + \text{OH} \rightarrow \text{CO}_2 + \text{H},
\]

if there is available OH. This oxidation mechanism is fast over a broad temperature range. At lower temperatures a slower reaction takes place:

\[
\text{CO} + \text{H}_2\text{O} \rightarrow \text{CO}_2 + \text{H}_2,
\]
The minimum CO emission can be found at slightly lean conditions, and the addition of oxygen increase its concentration. Besides $\lambda$, the operating pressure, the inlet temperature, the wall cooling (e.g., in gas turbine combustion chambers), and atomization have also an impact on this pollutant [2].

The reaction kinetics of unburnt or total hydrocarbons formation is more complicated than that of CO. While the governing factors of them found to be similar, as shown in Fig. 2.9, they are usually handled together. The differences arise mainly from low-temperature combustion (i.e., $< 1000 \text{ K}$) and inadequate droplet evaporation.

Most of the nitric oxide (NO) formed during combustion subsequently oxidizes to NO$_2$. For this reason, it is customary to lump NO and NO$_2$ together and express results in terms of NO$_X$, rather than NO [2]. This pollutant can be produced by five different mechanisms: thermal NO, prompt NO, NO$_X$ via N$_2$O, NO$_X$ via NNH, and fuel NO [99]. Considering lean combustion, the majority of the NO$_X$ formed through the thermal pathway. Note that the fuel NO is mostly related to coal combustion and all the investigated liquid fuels here are free from fuel-bonded nitrogen. The three governing reactions of thermal NO are:

$$\text{O} + \text{N}_2 = \text{NO} + \text{N}, \quad (2.30)$$

$$\text{N} + \text{O}_2 = \text{NO} + \text{O}, \quad (2.31)$$

$$\text{N} + \text{OH} = \text{NO} + \text{H}. \quad (2.32)$$

The required flame temperature is $> 1900 \text{ K}$ while the activation energy of Eq. 2.30 is high (314 kJ/mol) [99]. By reaching flame temperatures beyond 1900 K, the NO$_X$ concentration increases exponentially. As a conclusion, a slight disturbance in the combustion chamber
temperature profile progressively increase the NOX emission [66]. On the other hand, too low temperature decrease the overall efficiency of heat engines. Equations 2.30–2.32 also require residence time while thermal NO formation is a slow mechanism. Therefore, they are typical post-flame reactions. The primary reason to operate lean regarding NO emission is that the activation energy of the main step of prompt NO is significantly less (58 kJ/mol):

$$\text{CH} + \text{N}_2 = \text{NCN} + \text{H}.$$  

This reaction is important at as low temperatures as 1000 K. To lower the NO formation through this pathway, the CH concentration has to be reduced. Hence, excess oxygen is required that reacts fast with the CH radicals. At elevated pressures, the formation of NOX is increased, but it seems that above $\lambda = 1.5$, the difference is negligible in gas turbine combustion [66]. Moreover, it is not affected by the turbulence [66] unlike the emission of CO and unburnt hydrocarbons [4].

The soot emission is critical in fuel rich combustion applications. Its formation is not fully understood yet, but a possible pathway is the following:

$$\text{CH} + \text{C}_2\text{H}_2 = \text{H}_2\text{CCCH},$$

$$2\text{H}_2\text{CCCH} = \text{benzene},$$

$$\text{benzene} \rightarrow \text{PAH},$$

where PAH denotes the polycyclic aromatic hydrocarbons. The presented model of PAH formation is called Hydrogen abstraction and ACetylene Addition (HACA mechanism). The recombination of these molecules leads to the formation of soot particles. Note that the contribution of soot was underestimated to the climate change [99]. Nevertheless, lean premixed combustion counters the soot formation through the relatively low concentration of CH and C2H2 in the flame. Therefore, it is neglected in the flue gas analysis.

The last notable pollutant is the SOX. While it is typical in coal and Chinese mineral oil, it is also neglected in the present thesis. The pollutant emission analyzer was equipped with an SOX cell; it measured zero concentration at all cases. Nevertheless, the device was checked for this type of calibration gas as well. Typically, renewable fuels are free from sulfur.

The literature identifies three primary methods to achieve lower emissions. These are the following:

1. The simplest one is adding external fluid to the system (e.g., steam, water, N2, CO2). It is called wet emission control because the most commonly added fluid is water or steam. Historically, the first patent of water addition for combustion control dates back to 1791 [18]. If the concentration of H2O is high in the reaction zone, the atomic oxygen level significantly decreases [100]. Hence, the thermal NO formation rate is reduced through Eq. 2.30. If the water is emulsified with the fuel, the resulting micro-explosions due to the lower boiling point of water significantly improves the atomization quality [18]. However, quenching of CO burnout and facilitating hot corrosion of the turbine blades makes this technique less attractive [4].

22
2. To control the flame temperature, hence, the formation of NO\(_X\), staged (or rich-burn/quick-quench/lean-burn, RQL) combustion is used mainly in furnaces, boilers, and stationary gas turbines. In this case, the mixture is rich near the burner, characterized by increased CO and negligible NO\(_X\) formation. In order to complete the combustion, secondary air is introduced where the mixture becomes lean and the CO burns out. This method requires considerable residence time; therefore, it is uncommon in other combustion systems.

3. Using a well-designed flow pattern provide optimal \(\lambda\) at the flame front. Burner systems of this type are commonly called as Dry-Low-Emission (DLE) solutions [2] and used extensively in gas turbines. Even internal combustion engine designers follow the primary design considerations of DLE, but they refer to as swirl control. The DLE technique was developed primarily to lower the NO\(_X\) emission [4, 18]. Therefore, the term Dry-Low-NO\(_X\) (DLN), is also used in the literature but less frequently. The reason behind that is this design method helps in keeping the concentration of other pollutants low as well.

Apparently, all the low emission solutions must be further synchronized with the fluid dynamic properties of the combustion chamber to maintain a stable flame front across the entire operating range [65]. A common feature of all the DLE combustion chambers described above is that positive efforts are made to eliminate local regions of high temperature within the flame by mixing the fuel and air upstream of the combustion zone. While the formation of NO\(_X\) decreases with the increase of \(\lambda\), the more lean the mixture is, the better in this regard [2]. Only the flame stability sets a limitation. The name of this burner concept is Lean Premixed Prevaporized (LPP). It consists of three distinct regions. The first one is the fuel injection, which is an airblast atomizer in the present case. Then evaporation occurs which is followed by the mixing of combustion air and the fuel vapor. Hence, the burning droplets are eliminated, and the combustion is homogeneous and results in an even temperature field. Moreover, the soot is practically absent. Therefore, the flame is less luminous, and the liner receives less heat through radiation. The superiority of the LPP technology is supported by the fact that the gas and fluid fueled Capstone micro gas turbines only differ in their injection system. Otherwise, they have identical combustion chambers, which are characterized by similar pollutant emissions. In this way, the flame temperature can be easily kept below 1900 K that ensures low NO\(_X\) formation even at high residence times.

In order to control the combustion process and ensure low pollutant emission, exhaust gas analyzers seem to be the obvious choice. Depending on their measurement principles, they require few times ten seconds. If the loading varies dynamically or the system operates near the blowout limit, a low-response sensor is required. These technologies receive increasing attention due to the continuously stricter standards. Here the optical devices come into view, detailed in the upcoming section.

2.5 Flame diagnostic techniques

It was mentioned in Subsection 2.4.3 that being in line with the pollutant emission standards is mandatory. While the local equivalence ratio of the burner has the greatest impact on the pollutant emissions, shown in Fig. 2.9, its feedback control allows wide operating limits for combustion systems. Therefore, real-time monitoring requires measurement data within a short time to allow efficient control. The optical sensors are
non-intrusive, fast-response tools, and have low maintenance cost. Their operating principle is based on the detection of chemiluminescence emission of an intermediate radical (R) during combustion. In the first step, an excited radical is formed from a chemical reaction. A and B notes the parent species in Eq. 2.33 (the capital, non-italic letters mean that they are not variables). \( R^* \) loses its excess energy via spontaneous emission or collision with another species by emitting one photon (Eq. 2.34).

\[
A + B \rightarrow R^* + \text{other} \tag{2.33}
\]

\[
R^* (+M) \rightarrow R + h\nu (+M) \tag{2.34}
\]

At laboratory scales, Planar Laser-Induced Fluorescence is commonly applied for flame diagnostics. It is often combined with Particle Image Velocimetry to investigate the flame-vortex-acoustic interactions, see, e.g., [67,101–108]. However, practical problems (such as condensation and corrosion) make the application of these methods difficult in a harsh environment. For temperature mapping, pyrometry is a potential tool. Nevertheless, it relies on the thermal radiation of soot particles. Therefore, its application is limited to boilers and furnaces. Other modern combustion systems apply a less luminous lean flame, making such monitoring technique inappropriate [109]. A cost-effective solution is the chemiluminescence imaging through appropriate filtering. The measuring device can be even a commercial digital camera. Nevertheless, this technique falls at the point, where pyrometry is strong. The presence of soot or other broadband noise confines the application of chemiluminescence imaging to lean flames. Consequently, all the above-mentioned techniques are rather limited [109].

To date, the state-of-the-art development of online flame monitoring methods focuses on the use of flame emission spectroscopy, FES. They are characterized by low cost and robustness [60,61,109–117]. The theoretical framework of FES for equivalence ratio monitoring was developed for laminar premixed flames, especially for methane-air combustion [60,61,114–117]. Despite this fact, it is also applicable to turbulent flames at industrial scales [109–111]. Only few papers discuss FES of liquid fuel combustion which are mostly related to reciprocating engines [118–120]. The investigations of non-premixed flames (see, e.g., [121,122]) are generally out of scope due to their corresponding higher pollutant emission in practical scales. The spectrometry of crude vegetable oil combustion is absent in the literature according to the best knowledge of the author. Consequently, the present thesis aims to investigate a LPP burner utilizing both standard diesel and crude rapeseed oils. Therefore, application of FES is discussed in Subsection 2.5.1.

### 2.5.1 Flame emission spectrometry

As it was mentioned above, FES is capable of measuring the local \( \lambda \). For this purpose, chemiluminescence emission of excited radicals, such as \( \text{OH}^* \), \( \text{CH}^* \), \( \text{CO}_2^* \), \( \text{C}_2^* \), or an entire spectrum in the UV–VIS range is measured [109,114,117]. Mostly the \( \text{CH}^*/\text{OH}^* \) is used for determining the local equivalence ratio since it shows a power law relation with both fuel-to-air equivalence ratio and operational pressure [60,61,110]. The mass flow rate of the premixed fuel was found to be proportional to the chemiluminescence intensity [60,61,110]. Therefore, it is less important to test a burner at different loadings at the same \( \lambda \), so it is neglected in the presented measurement series in Chapter 6. The most important reactions which play a significant role in the formation of excited radicals are excellently
reviewed by Panoutsos et al. [123]. They also found that OH* closely follows the trend of heat release rate.

Docquier et al. [117] concluded that the CH*/OH* chemiluminescence intensity ratio of methane-air flame is measurable even at 20 bar operating pressure. Chemiluminescence intensity of C2* at 516 nm (further noted as C2*516) under ambient conditions weakens with increasing pressure. Furthermore, it faded to the background noise above 10 bar, which makes the control of the flame cumbersome based on the chemiluminescence intensity at such high pressures. Small-scale gas turbines typically operate below the pressure ratio of 5. Consequently, C2*516 chemiluminescence intensity seems to be feasible for diagnostics and active control purposes for such heat engines as well. The list of potential applications contains all steady-operating combustion systems. Note that Muruganandum et al. [113] pointed out that the presence of unevaporated fuel droplets can affect the chemiluminescence intensity ratio measurement.

As it was mentioned above, the present thesis focuses on the measurement of diesel and crude rapeseed oil combustion. As the technical literature contains no information about chemiluminescence analysis of crude vegetable oils, it is a pioneer work, discussed in Section 6.4.
Chapter 3
Atomization

While atomization is the first step in liquid fuel combustion, it is crucial to know various parameters, such as droplet motion, average size, and distribution to properly design a combustion chamber. Therefore, the present chapter discusses these properties of a spray generated by an airblast atomizer. The role of the dynamical analysis is to analyze the spray structure with special attention on the Stokes number, the turbulent kinetic energy, and the change of droplet velocities besides determining the downstream distance where the spray can be considered as fully developed. While there are numerous formulas available in the literature to estimate the Sauter Mean Diameter, these are fitted to the measurement results to find out which ones describe the atomization process properly in the investigated high-velocity region. Besides the $SMD$, the size distribution of the spray is also relevant [13]. However, this aspect of atomization is rarely investigated [40].

Figure 3.1 shows the cross section of the investigated atomizer. Note that the mixing tube (shown in Fig. 4.1) was removed in order to isolate the atomization. Other researchers followed the same way in case of this burner [32–34]. The fuel discharges from a 0.4 mm diameter central pipe. Concentric with that, the high-velocity atomizing air flows around the liquid core and accelerates the fuel stream from an annular nozzle with inner and outer diameter of 0.8 and 1.6 mm, respectively. Thus, the fuel jet is shattered into small fractions due to the liquid-air interactions. Note that the present burner was designed for the Capstone C-30 micro gas turbine, operating on diesel oil or kerosene.

Regardless that other chapters investigate both diesel and rapeseed oils, this one is confined to diesel oil only. Unfortunately, there was no opportunity to carry out the below-presented measurement series using rapeseed oil. Therefore, it is a highly potential future direction of this work.
3.1 Phase Doppler Anemometry measurement setup

The problem of measuring large number of small particles is present in many fields of engineering. However, spray measurement is one of the most challenging aspects, while the number of droplets is very large, their velocity is high and varying, the size of the largest to the smallest droplet may exceed 100:1. The droplet sizes may change during the measurement due to evaporation and coalescence. Therefore, the following requirements have to be met:

- Create no disturbance to the atomization process.
- Able to capture such small droplets as 1 µm in diameter.
- Measure the spray both spatially and temporary.
- Measuring at least 5000 samples is recommended for good statistical representation.
- Tolerate wide variations in the liquid and the ambient gas properties.
- High data rate.

These requirements were suggested by Lefebvre in 1989 [13], but fulfilling all conditions is challenging even today. Optical approaches overcame the old technologies, such as frozen-drop, drop collection slide, and hot-wire measurements. Imaging techniques provide valuable information, but image processing always requires human involvement even today [6]. Consequently, their purpose is often qualitative rather than quantitative analysis of spray formation.

Among non-imaging optical techniques, the light scattering method applied most extensively for droplet sizing in the range of millimeters to a hundred nanometers, like the
PDA. The measurement setup is detailed in Fig. 3.2. All the tests presented in this chapter were carried out at so-called cold conditions, so the fuel was not ignited. The used cylindrical coordinate system is indicated at the nozzle discharge.

The desired 0.35 g/s diesel oil mass flow rate was achieved by using a pressurized fuel tank at 2.5 bar and a control valve was used to set the desired value. The fuel properties were the following: $\nu = 3.5 \text{ mm}^2/\text{s}$, $\rho = 825 \text{ kg/m}^3$, and $\sigma = 0.025 \text{ N/m}$. The atomizing air was taken from a 6 bar compressed air system through a pressure regulator valve. Both lines were equipped with mass flow meters, thermocouples, and pressure transducers. The generated spray was collected into an open tank and the remaining fuel mist was vented out by a fan through filters. The following atomizing pressures were investigated: $p_g = 0.3, 0.5, 0.7, 0.9, 1.1, 1.6, 2.1$, and $3.1 \text{ bar}$.

Figure 3.3 shows a schematic drawing of the 2D Fiber based PDA system made by Dantec Dynamics, emphasizing the optical setup. The scattering angle was 70 $^\circ$, which is the so-called Brewster’s angle, characterized by one dominant light scattering mode. The focal lengths were 500 mm for both the transmitting and the receiving optics.

A series of measurement points are indicated in Fig. 3.3 in a horizontal section. Due to the preliminary investigation showed that the generated spray is symmetric, selected points on a diameter were measured at each atomizing pressure. In order to analyze the evolution of the spray, four downstream distances were investigated. Due to the velocity measurement limitations ($\sim 300 \text{ m/s}$), the closest one was at 10 mm downstream from the nozzle discharge. The next one was at $z = 15 \text{ mm}$, which was followed by $z = 26.7 \text{ mm}$. The latter one was chosen as a typical distance, while it is the inner diameter of the mixing tube – which was removed. The last axial distance was $z = 50 \text{ mm}$, while the spray found to be developed at this distance, using the Weber number-based stability criterion, Eq. 2.20. The results are discussed in Section 3.3.

The number of sampling points through a diameter was thirteen, except $z = 50 \text{ mm}$, where it was fifteen. The radial step was 1 mm at $z = 10$ and $15 \text{ mm}$, while it was 2 mm at $z = 26.7$ and $50 \text{ mm}$. The optical system was kept at a fixed position while an automatized traverse system moved the atomizer step by step with 0.1 mm positioning uncertainty. Further details of the measurement configuration with the respective uncertainties are summarized in Appendix A. A single measurement point took 20,000 samples or 15 s. The automatized traverse system moved to the next position after fulfilling either condition.
The latter limitation was made due to the peripheral regions, where only few hundred droplets were measured.

The evaporation of the spray might influence the results considerably. Therefore, this effect was investigated by using the $D^2$-law, detailed in Section 2.3. Assuming that the highest residence time of a single particle among the currently investigated operating conditions, it turned out that only $D < 0.161 \mu$m droplets may evaporate completely in the worst case. However, it is recommended to use the droplet size that requires at least ten times larger residence time than it is available to ensure that the effect of evaporation can be neglected [124]. In this case, it is $D > 0.51 \mu$m which refers to 99.8% of the measured droplets by number fraction. Consequently, it is assumed in this chapter that the spray evaporation does not affect the measurement procedure. The evaporation of droplets was calculated by Eqs. 2.21–2.28.

### 3.2 Droplet dynamics

![Figure 3.4: Size-velocity correlations at $p_g = 0.9$ bar and various axial distances: a) $z = 10$ mm, b) $z = 15$ mm, c) $z = 26.7$ mm, and d) $z = 50$ mm. All results were obtained at $r = 0$ mm.](image)

The liquid is disrupted by the turbulent behavior of the flowing gas and the newly created liquid fractions are further accelerated which leads to the formation of ligaments; these then break up into smaller droplets [46]. Figure 3.4 shows a typical radial-axial velocity scatter plot at $p_g = 0.9$ bar for all four axial distances, indicating the spray development. Estimation of the maximum stable droplet diameter, defined by Eq. 2.20, shows that majority of droplets should not breakup further even in the critical near-nozzle region. At $z = 10$ mm from the nozzle, the maximal stable diameter fits within the range...
of 10 to 115 µm (not shown here), which suggests that only a small fraction of measured particles might undergo a breakup in the center. The high turbulent intensity causes the significant scatter of the droplet velocities. The derived turbulence characteristics will be shown later in Fig. 3.6.

The near-nozzle region \((z = 10 \text{ mm})\) is characterized by a broad spectrum of radial droplet velocity. \(D < 20 \mu \text{m}\) particles reach higher radial velocities than the larger ones, due to the highly turbulent atomizing jet [46], shown in Fig. 3.4a. It is supported by the fact that the majority of \(D < 10 \mu \text{m}\) droplets have equal or higher axial velocity than that of \(D > 20 \mu \text{m}\) particles in the vicinity of the nozzle. The velocity of small particles is significantly reduced as the spray evolves, following the expectations. Interestingly, Fig. 3.4d \((z = 50 \text{ mm})\) shows that the large droplets keep their momentum, hence; the velocity, which is in a good agreement with the literature [124]. When comparing Figs. 3.4a–d, the transition of droplet distribution is revealed. In the near-nozzle region, high-velocity droplets are generally smaller than 15 µm. However, as the spray evolves, the droplets decelerate due to their interaction with ambient air.

![Figure 3.5: Size-velocity correlations at \(z = 50 \text{ m}\) and various atomizing pressures: a) \(p_g = 0.3 \text{ bar}\), b) \(p_g = 0.9 \text{ bar}\), c) \(p_g = 2.1 \text{ bar}\), and d) \(p_g = 3.1 \text{ bar}\). All results were obtained at \(r = 0 \text{ mm}\).](image)

Figure 3.5 shows scatter plots at \(p_g = 0.3, 0.9, 2.1,\) and 3.1 bar, respectively. The low-pressure regimes are characterized by a strong axial flow while the radial component remains relatively weak, which is caused by low turbulence intensity. With an increase of pressure, the velocity grows in both axial and radial direction together with the turbulence.

While the following part will present the energetic structure of the spray, besides the TKE, defined in Eq. 2.19, the mean kinetic energy, \(MKE\), was also calculated as follows:

\[
MKE = 0.5 \left[ (\overline{w_z})^2 + (\overline{w_r})^2 + (\overline{w_\phi})^2 \right],
\]

(3.1)
where overbars indicate the ensemble average, $z$, $r$, and $\varphi$ are the axial, radial, and tangential components of the velocity, respectively. In order to distinguish the liquid and the gas phase, the $D \leq 3 \mu m$ droplets were filtered and averaged. In order to determine what extent the droplets follow the streamlines, the Stokes number, $Stk$, was determined as follows:

$$Stk = \frac{(\rho_L - \rho_A) D (w_A - w_L)}{18\mu_L},$$ (3.2)

where $\rho$ is the density, $D$ is the diameter of the droplet, $w$ is the velocity, $\mu$ is the dynamic viscosity, and subscripts $A$ and $L$ refer to the air and liquid phase, respectively. While the largest $Stk$ number was below 0.01 at $p_g = 0.3$ bar $z = 10$ mm, and $r = 0$, it can be assumed that the droplets follow the streamlines well. $Stk < 10^{-4}$ at $z = 50$ mm at all radial distances and atomizing pressures.

Figure 3.6: Radial profiles of TKE, MKE, axial and radial velocity profiles. All of these characteristics are shown for liquid and gas phase in the spray and for: a) $z = 10$ mm, $p_g = 0.9$ bar, b) $z = 15$ mm, $p_g = 0.9$ bar, c) $z = 26.7$ mm, $p_g = 0.9$ bar, d) $z = 50$ mm, $p_g = 0.9$ bar, e) $z = 50$ mm, $p_g = 0.3$ bar, f) $z = 50$ mm, $p_g = 0.9$ bar, g) $z = 50$ mm, $p_g = 2.1$ bar, h) $z = 50$ mm, $p_g = 3.1$ bar. Note the different radial scale in a) and b).

Figure 3.6 shows the energetic structure of the spray through separating the gaseous and the liquid phase. It quantitatively reveals the decay of the two-phase jet and presents the development of its core which incorporates intense atomization. The following physical quantities are shown: TKE, MKE, axial and radial velocity profiles at various axial
distances at $p_g = 0.9$ bar, a-d, and at different atomizing pressures at $z = 50$ mm, e-h. Note that Fig. 3.6d and f are the same results but in a different scale. This is for easier comparison of the trends in the two columns. The relative radial position here and Figs. 3.8 and 3.9 means that the measured values in both directions were averaged based on their distance from the origin.

There is a clear trend of decay in the axial and radial velocities for both the liquid and the gas phase with growing axial distances. By separating the motion of the two phases, the overshooting phenomenon, explained in the literature [46], is clearly observable. Namely, the near-nozzle region is characterized by acceleration of the larger droplets which dissipate their kinetic energy slower than the gas phase, confirmed by both the velocity and the $MKE$ trends.

When examining the $TKE$ and the $MKE$ profiles concerning the operating regime, it can be seen that their values are directly proportional to the atomizing pressure mainly at the center of the spray. The $TKE$ profiles are very similar in both liquid and gas phases. For atomizing pressure of 0.3 bar, the maximum value of $TKE$ is concentrated in the center of the spray. However, when the pressure increases the maximum value moves radially from the center to $r = 2$ mm, due to the intensifying shear between the liquid and the gas phases. $MKE$ profiles show that the energy is concentrated mainly in the liquid phase in the vicinity of the axis. The difference between $MKE$ of liquid and gas phase decreases with the growth of the atomizing pressure. It points to the fact that the kinetic energy is transferred from the gas phase to the liquid more intensively in the high-pressure regimes which are characterized by higher $TKE$s.

Figure 3.7 shows the influence of the operating gas pressure on the relation between the droplet size and the axial velocity, which is a dominant component in the examined spray. These results were obtained at $z = 50$ mm and $r = 0$ mm. The overshooting phenomenon is also confirmed by Fig. 3.7 while the average velocity of the droplets increases with their size. Moreover, the slope of the nearly linear profiles increases with the atomizing pressure. It can be explained by the fact that the discharge velocity increases with the atomizing pressure. As expected, the droplet size is negatively correlated with the operating pressure. With the growth of atomizing pressure, droplets are smaller, and the velocity fluctuations enlarge in parallel with the $TKE$.

![Figure 3.7: Size-velocity correlation at $z = 50$ mm and $r = 0$ mm a) $p_g = 0.3$ bar and b) $p_g = 0.3, 0.9, 2.1,$ and 3.1 bar.](image-url)
3.3 Global spray characterization

Figure 3.8 shows the evolution of the spray at four atomizing pressures and all the four axial distances. At $p_g = 0.3$ and 0.9 bar, such a significant peak at the center is absent due to the lower discharge velocities. Here, the fuel jet has more time to break up, hence, a more evolved spray is measured at $z = 10$ and 15 mm.

![Figure 3.8: Radial SMD distribution of the spray at a) $p_g = 0.3$ bar, b) $p_g = 0.9$ bar, c) $p_g = 2.1$ bar, and d) $p_g = 3.1$ bar.](image)

Based on the results, a nearly complete state of the spray was measured at these atomizing pressures. The spray expands, but slight changes can be observed regarding the SMD values in the downstream regions. The spray needs more axial distance to develop fully at high atomizing pressures, indicated by the data of $p_g = 2.1$ and 3.1 bar. The central region clearly indicated the ongoing secondary atomization. Typically, the droplets drifted to the periphery do not undergo a secondary breakup; however, they represent only a small portion of droplets. Nevertheless, these droplets may considerably influence the combustion efficiency and pollutant emission of a burner. It occurs when unevaporated droplets or highly heterogeneous fuel-air mixture is present at the flame front. Therefore, high atomizing pressure ensures not only smaller droplets but also a more even spray. It should be kept in mind that the increased atomizing pressure is available at the expense of lower atomization efficiency.

![Figure 3.9: Comparison of the a) calculated stable droplet sizes with b) SMD at $z = 50$ mm.](image)
Figure 3.9 shows the droplet stability analysis at \( z = 50 \text{ mm} \) using Eq. 2.20 and comparing the results with the derived SMD. The figure confirms that the peripheral droplets of larger sizes remain stable while the most intense droplet breakup takes place at the center.

At high atomizing pressures, the stable droplet sizes at the axis are close to the SMD values determined from the measurements. However, at \( p_g = 3.1 \text{ bar} \), the stable droplet size (7.95 \( \mu \text{m} \)) is slightly larger than the SMD (8.25 \( \mu \text{m} \)). As a consequence, the spray can be considered as fully developed at \( z = 50 \text{ mm} \). It should be noted that such a stability analysis close to the nozzle is cumbersome at high atomizing pressures, due to the high turbulence intensity while Eq. 2.20 considers it through an averaged parameter, \( \varepsilon \).

### 3.4 Validation of SMD estimating formulas

During a pre-design, the temporal, spatial, and breakup properties of the liquid jet are not necessarily required. To characterize a droplet size distribution in the spray with a single value, the integral SMD (ISMD) is calculated as follows [125]:

\[
\text{ISMD} = \frac{\sum_{i=2}^{n} r_i D_{30,i}^3 f_i}{\sum_{i=2}^{n} r_i D_{20,i}^2 f_i},
\]

where the subscripts of \( D \) refer to Eq. 2.5, \( f \) is the data rate in Hz, and \( r \) denotes the radial coordinates in which the measurements were carried out. Moreover, this term is responsible for the appropriate weighting of the results since the spray is axisymmetrical. The sum begins from \( i = 2 \) as the formula excludes an inner circular area, which is a single point mathematically and does not influence the results. Figure 3.10 shows the ISMD for all operating regimes and all measured axial distances.

![Figure 3.10: ISMD of the spray at various axial distances.](image)

Interestingly, at \( z = 10 \text{ mm} \), high atomizing pressures probably lead to larger measured droplet sizes due to the high discharge velocities, showing that larger distance is needed for the complete breakup of the unstable droplets. This phenomenon is also observable in Fig. 3.8. Namely, the fully developed spray at \( z = 50 \text{ mm} \) is characterized by lower ISMD values by increasing the atomizing pressure. When reaching the critical pressure ratio at \( p_g = 0.89 \text{ bar} \), hence, achieving supersonic flow at the nozzle discharge, there is an apparent break in the ISMD trends at the first three downstream distances. It suggests a
slight interaction between the spray and the emerging shock waves, which is in agreement with the literature [126]. Regardless that visibly the spray seems to be developed yet at \( z = 26.7 \) mm, Figs. 3.8 and 3.9 show that atomization still occurs there which ends only at \( z = 50 \) mm.

The upcoming part of the section is a validation check of Eqs. 2.6–2.11. It is known that the constants may depend on the design [13] but the present investigation aims to figure out which ones describe accurately the atomization process. However, this idea is not new, Nakamura et al. [33] only referred to this method and not considered it in their paper. One of their students, Bolszo suggested in 2005 that the famous formula of Lefebvre, Eq. 2.7, may not be accurate enough under all conditions [32]. This finding is also supported by Krishna [34]. An identical atomizer to that of Fig. 3.1 was used in all the cited papers. More recently, Chong and Hochgreb published a paper [127] in which they investigated the validity of Eqs. 2.7 and 2.8 using an internal mixing airblast atomizer. They concluded that Eq. 2.7 works well while the latter one is less accurate. However, the exponent of \( \rho_L \) in Eq. 2.8 was 0.87 instead of 0.37 in their paper. It is not known whether it is a typo or a source of miscalculation which affects their results, hence, the conclusions. The most significant discrepancy of their study is that the calculated discharge velocity of air did not consider the further expansion downstream the orifice. It makes the mentioned results highly doubtful. While this recent paper focuses on the validation of \( SMD \) estimating formulas, it indicates that this field is still highly relevant today. Figure 3.11 shows the fitted \( SMD \) estimations to the data set at \( z = 50 \) mm.

Table 3.1 shows the used constants and the resulting \( R^2 \) values of Fig. 3.11. Equation 2.6 showed the second best fit at \( A = 0.61 \) and \( B = 0.041 \). Therefore, this equation was tuned further to eliminate the viscosity term as it is expected to have low influence in the case of high-velocity atomization [20,36,37]. In this case, \( A = 0.66 \) resulted in the best fit while \( B \) was zero. The coefficient of determination decreased slightly, but Eq. 2.6 became very simple. It depends now only on \( \dot{W}_{eA} \) and \( ALR \).

Eq. 2.7 with the original constants (denoted as Eq. 2.7 orig. in Fig. 3.11) remarkably overestimated the \( SMD \) in the investigated region. By varying the constants, it turned out that the best fit can be achieved at \( E = 0 \), suggesting that prompt atomization takes place here. Here, \( C = 0.47 \) showed the best fit which is very close to the original value of this constant [31]. Equation 2.9, derived by Varga et al. [36], showed an excellent fit to the measurement data, resulting in \( R^2 = 0.991 \) at \( I = 0.297 \sqrt{m} \). They suggested
Table 3.1: Summary of the fitted equations of Fig. 3.11.

<table>
<thead>
<tr>
<th>Equation</th>
<th>Author</th>
<th>Constant 1</th>
<th>Constant 2</th>
<th>$R^2$</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.6</td>
<td>Lefebvre [12]</td>
<td>0.61</td>
<td>0.041</td>
<td>0.997</td>
<td></td>
</tr>
<tr>
<td>2.6</td>
<td>Lefebvre [12]</td>
<td>0.66</td>
<td>0</td>
<td>0.986</td>
<td>neglecting the viscosity term with the original constants</td>
</tr>
<tr>
<td>2.7</td>
<td>Rizk &amp; Lefebvre [31]</td>
<td>0.48</td>
<td>0.15</td>
<td>0.0929</td>
<td></td>
</tr>
<tr>
<td>2.7</td>
<td>Rizk &amp; Lefebvre [31]</td>
<td>0.47</td>
<td>0</td>
<td>0.926</td>
<td></td>
</tr>
<tr>
<td>2.8</td>
<td>Lorenzetto &amp; Lefebvre [35]</td>
<td>0.39</td>
<td>0.13</td>
<td>0.998</td>
<td></td>
</tr>
<tr>
<td>2.9</td>
<td>Varga et al. [36]</td>
<td>0.297 $\sqrt{m}$</td>
<td>-</td>
<td>0.991</td>
<td></td>
</tr>
<tr>
<td>2.10</td>
<td>Lefebvre [37]</td>
<td>0.00082</td>
<td>-</td>
<td>0.702</td>
<td></td>
</tr>
<tr>
<td>2.11</td>
<td>Park et al. [20]</td>
<td>2.04</td>
<td>-</td>
<td>0.0418</td>
<td></td>
</tr>
</tbody>
</table>

$I = 0.055 \sqrt{m}$ based on their measurement results. It is 5.4 times less regardless that their atomizer configuration was very similar. While Eq. 2.9 does not contain the diameter of the liquid jet, a significant variation can be addressed to the different geometry or considerably higher discharge velocities ($w_A = 208 - 445$ m/s assuming adiabatic expansion at $p_g = 0.3 - 3.1$ bar). Therefore, more investigations are needed; this might help in finding the physical meaning of $I$. There is no doubt that Eq. 2.9 has the most solid physical background, but it turned out that the constant significantly depends on the atomizer design. Hence, the use of Eq. 2.6 or 2.7 might result in a better prediction. The formula of Lorenzetto and Lefebvre was the most accurate one, however, only the first constant was decreased to 0.39 from the originally suggested 0.95 [35]. It was a surprise that the coefficient of the viscosity term showed the best fit at its original value. Regardless of the $R^2 = 0.998$, which is the best one, this formula is not suggested to predict the $SMD$ while it contradicts with all the previous findings that the role of viscosity vanishes at high-velocity atomization.

Equation 2.10 predicted the $SMD$ less accurately. While increased $p_g$ results in a lower atomization efficiency, it advises that $G$ should not be a constant while it is the modified atomization efficiency. Therefore, the suggested equation by Lefebvre [37], which was derived by solid theoretical assumptions, was significantly outperformed by Eqs. 2.6–2.9. The fit of Eq. 2.11, suggested by Park et al. [20], showed the worst fit with $R^2 = 0.0418$ at $K = 2.04$ even though the atomization efficiency is the function of $p_g$. This was not expected while Eqs. 2.11 and 2.12 were derived under supersonic conditions. Namely, the validity is in the range of $p_g = 1 - 4$ bar, which greatly overlaps our investigated atomizing pressure regime. By considering that the ambient pressure negligibly affects $SMD$ [23], the validation of Eqs. 2.6–2.11 was carried out at Mach number, $Ma = w_A/c = 0.6 - 1.6$. Here, $c$ is the speed of sound, and the calculated values are based on the measurement conditions, assuming adiabatic expansion.

### 3.5 Droplet size distribution

Figure 3.12 shows the parameters of the $\Gamma$, the Rosin-Rammler, RR, and the Nukiyama-Tanasawa, NT, PDFs by curve fitting method. Measurement points with insufficient data (i.e. less than 20,000 samples in 15 s) were neglected to achieve more correct fits.
These peripheral regions showed varying and irregular droplet distributions. Practically, it means that \(-10 \text{ mm} \leq r \leq 10 \text{ mm}\) regime was taken into account at all atomizing pressures.

Figure 3.12: Average parameters of the a) \(\Gamma\), RR, and b) NT PDFs at \(z = 50\) mm.

Figure 3.13: Droplet size distributions at various atomizing pressure, using \(\Gamma\) PDF.

It should be noted that the four-parameter NT distribution function was also analyzed, but the excessive change of the parameters lead to the exclusion of that function. Interestingly, the best fit was achieved by the \(\Gamma\) function with \(R^2 = 0.983\) on average, outperforming both RR \((R^2 = 0.962)\) and NT \((R^2 = 0.982)\) PDFs, shown in Fig. 3.14.
The value of $R^2$ increases with the atomizing pressure as the spray becomes more homogeneous. The most significant theoretical discrepancy of the NT function based on the averaged parameters is that the integral of these PDFs was not equal to unity, shown in Fig. 3.15. At $p_g = 2.1$ bar this value was decreased to 0.65. Nevertheless, both $\Gamma$ and RR PDFs fulfill this condition in the whole range. Nevertheless, at 0.3 and 0.5 bar atomizing pressure, the integral of the NT PDF was above 0.9, making it an optional choice in this region. Furthermore, in this region, NT shows a better fit to the droplet distributions while it is able to incorporate a wider range of droplet sizes, which is the situation at low atomizing pressures. Therefore, the decreased $R^2$ values of NT PDF are expected in more homogeneous sprays.

Based on the measurement results, the $\Gamma$ PDF is recommended for describing the size distribution of sprays generated by high-velocity air stream above RR and NT PDFs. Currently, Fig. 3.12 can be used to estimate the size distribution of atmospheric diesel oil atomization in the $p_g = 0.3 - 3.1$ bar range. However, it requires future work to find more general trends and derive formulas for the PDF estimation besides the numerous SMD formulas.
3.6 Summary of the chapter

The first section of the chapter detailed the measurement setup and the investigated conditions of the cold atomization test rig, assembled at Brno University of Technology. Section 3.2 showed the dynamic and energetic structure of the spray. Due to the low \( Stk \) numbers, the droplets followed the streamlines well and no significant difference was found when the variables were determined by droplet size filtering at 3 \( \mu m \). The smaller droplets were considered as they represent the motion of air and it was compared with the whole data set at selected atomizing pressures and all the four downstream distances. Here the turbulent and mean kinetic energies showed the energetic structure of the spray. Section 3.3 discussed the droplet stability analysis and spray evolution. Here, the measurement results were used to determine a single \( SMD \) value at each measurement point and it was compared with the calculated maximum stable droplet diameter at the respective condition, using Eq. 2.20.

The averaged \( SMDs \) at all the four axial distances were presented in Section 3.4. While \( z = 50 \) mm was found a proper downstream distance for having a fully developed spray, numerous empirical formulas available in the literature, discussed in Subsection 2.2.1, were fitted using the measurement data. It turned out that the role of the viscosity is negligible at high-velocity atomization. Only few formulas described the atomization process accurately. Among them, using Eq. 2.6 is recommended by neglecting the viscosity term. Nevertheless, it is not the most accurate formula but a highly recognized and a simple one, which can be used at high atomization pressures. Equation 2.9 has a solid physical background, however, a single term is missing here to eliminate the need for empirical constants. Therefore, it requires further research.

Lastly, three droplet size distribution functions were analyzed from which the \( \Gamma \) PDF performed most accurately. Due to its simplicity, it is highly recommended to use this function for characterizing droplet size distributions over the widely recognized Rosin-Rammler or the Nukiyama-Tanasawa PDFs. The following new scientific result was derived from the results of the chapter:

Based on the findings of Lefebvre [12], Heng et al. [23] and Varga et al. [36], the \( SMD \) negligibly depends on the ambient pressure and \( SMD \sim We_A^{-1/2} \) at high-velocity atomization. By neglecting the viscosity term in Eq. 2.6, its range of validity is significantly extend as follows:

1. Thesis

In the case of plain-jet airblast atomization of standard diesel oil, the following formula estimates the volume-to-surface diameter, \( SMD \), of the spray in the range of Mach number, \( Ma = 0.6 - 1.6 \) and air-to-liquid mass flow ratio, \( ALR = 0.78 - 2.3 \):

\[
SMD = 0.66d_0 We_A^{-0.5} (1 + 1/ALR),
\]

where \( d_0 \) is the diameter of the liquid jet and \( We_A \) is the Weber number based on the properties of the atomizing air.
Chapter 4

Evaporation

It was mentioned in Chapter 2 that the droplets should evaporate completely and mix with the combustion air before reaching the flame front. This requirement immediately turns into an optimization problem through the following phenomena. Firstly, small droplets, produced by a high-velocity atomizing are favorable, since their evaporation time is short. In this case, the residence time is short, too due to the significant influence of the atomizing jet. Secondly, a longer residence time through low atomizing velocity may not necessarily help the situation while the evaporation time grows in parallel to that through the larger droplet sizes. Furthermore, Eq. 2.7 suggests that the achievable droplet size is limited by the second term, containing the $Oh_L$ number, which is free from the relative velocity. In the present chapter Eq. 2.7 is used for SMD estimation over Eq. 3.4 while low-velocity atomization is also taken into account.

The investigated burner is shown in Fig. 4.1. Its atomizer was discussed earlier in Chapter 3. The mixing tube is 75.5 mm long and has a 26.8 mm inner diameter. The combustion air enters through four radial orifices and fifteen $45^\circ$ swirl vanes. The combustible mixture leaves the burner lip and ignites.

![Figure 4.1: The investigated LPP burner.](image)

Regarding the residence time, mentioned in the first paragraph, it is necessary to detail the role of the mixing tube. By design, the flow velocity is higher inside of it than the burning velocity, allowing the fuel droplets to evaporate properly and ignite after leaving the tube. This is the conception of the LPP burner, mentioned in Chapter 2. While the droplets are highly influenced by the atomizing jet, it receives a special emphasis in this chapter, leading to the calculation of the residence time of the atomized droplet. Next, the calculation of evaporation is discussed, based on the $D^2$ law, using Eqs. 2.21–2.28. Then the evaporation number is introduced which defines the residence-to-evaporation time ratio. This value is derived for various fuels versus $p_g$ in Section 4.1. In order to increase the vaporization rate, fuel and combustion air preheating were investigated in Section 4.2. Note that the the majority of the temperature-dependent material properties is listed in Appendix C. Since the $D^2$ law is universal and conservative rather than highly
accurate, the aim of the current analysis is to qualitatively compare the evaporation process utilizing various fuels.

4.1 Calculation of the evaporation number

Two assumptions are made in order to simplify the calculation method. Firstly, there is one-way coupling between the atomizing jet and the fuel jet. This one eliminates the inclusion of fuel flow rate through various loadings and the degree of mixing. The latter one is further affected by the intense turbulent motion of the droplets, shown in the previous chapter. Consequently the second assumption is that the droplets are traveling only axially. Hence, the problem is simplified to one dimension, surely underestimating the residence time in order to remain conservative. Obviously, there are available advanced calculation methods, but this one is enough for the desired qualitative comparison.

In order to determine the accurate mass flow rate of the atomizing air at a given atomizing gauge pressure, a measurement was carried out. The configuration, shown in Fig. 4.2, consisted a pressure regulator, which was connected to the high-pressure air system of the laboratory, then a rotameter, and a pressure transducer. The whole setup is placed on a scale.

![Figure 4.2: Test rig for thrust measurement.](image)

Hence, the thrust is calculated by multiplying the weight with the gravitational acceleration. While the volume flow rate is measured, the discharge velocity and the mass flow rate can be calculated.

Figure 4.3 shows the results with a fitted power law function, including the combined expanded uncertainties at 95% level of significance. The error bars of the pressure measurement is omitted, due to the high accuracy of the pressure transducer (0.25% of full scale, which is 1000 kPa here). Then the obtained results were compared to the adiabatic expansion of the atomizing air. However, there was no significant difference identified between them, the equation of the fitted curve is used for further calculations in this chapter. The difference in mass flow rate is increasing notably above 2 bar atomizing gauge pressure. It can be explained by the non-adiabatic process in the reality, i.e., the heat transfer between the environment and the jet is increasing with the atomizing pressure.

After the nozzle discharge, shown in Fig. 4.1, the atomizing air forms an annular free jet. As it was mentioned in the first paragraph of the present section, one-way coupling is assumed between the fuel and the atomizing jet. The following formula, suggested by Zawadzki et al. [128], was used for determining the velocity decay at the axis:
$w_A(z) = \begin{cases} w_A, \text{ if } z \leq 6, \\ 6w_A h_A/z, \text{ if } z > 6, \end{cases}$ (4.1)

where $h$ denotes the equivalent hydraulic diameter. Regardless that Eq. 4.1 was derived by using a unique measurement configuration, it shows excellent agreement with general free jet theories, while the core length of a conventional axisymmetric free jet, where the velocity is the same as at the discharge, is typically $5-6 h_{A,0}$ [78].

The following fuels were investigated from atomization point of view: ethanol, n-heptane, diesel oil, RME, jatropha, palm, rapeseed, soybean, and sunflower oils. Since the investigation is confined to subsonic atomization, Eq. 2.7 is used for droplet size calculation rather than Eq. 3.4 while the lower limit of the calculation was $p_g = 0.053$ bar.

In order to understand the difference between the aforementioned crude vegetable oils, their chemical structure is detailed below.

Vegetable oils are mostly composed of triglycerides, which contain a glycerol molecule and three fatty acids connected to it [9] which influences the physical properties of the fuel. The probability of each fatty acid in the case of different vegetable oils is shown in Table 4.1. In the second column, the number of carbon atoms and the number of unsaturated bonds are noted. If the probability of a fatty acid in a vegetable oil is about zero, it is marked with not relevant, NR. While the weather and the climate has a notable impact on the molecular composition, typical ranges were shown in Table 4.1. Consequently, the physical properties of the crude vegetable oils may also vary. Note that these fuels are all newtonian fluids [129].

<table>
<thead>
<tr>
<th>Fatty acids</th>
<th>Structure</th>
<th>Rapeseed</th>
<th>Sunflower</th>
<th>Palm</th>
<th>Soybean</th>
<th>Jatropha</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lauric</td>
<td>12:0</td>
<td>NR</td>
<td>0-0.5</td>
<td>0.13-0.23</td>
<td>NR</td>
<td>0.31</td>
</tr>
<tr>
<td>Myristic</td>
<td>14:0</td>
<td>&lt;0.1</td>
<td>0-0.2</td>
<td>0.85-0.91</td>
<td>&lt;0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Palmitic</td>
<td>16:0</td>
<td>3-4.7</td>
<td>3.5-6.7</td>
<td>36.75-40</td>
<td>11.2</td>
<td>13.4-14.2</td>
</tr>
<tr>
<td>Stearic</td>
<td>18:0</td>
<td>1-2</td>
<td>1.3-5.9</td>
<td>2.49</td>
<td>2.9</td>
<td>5.44-7</td>
</tr>
<tr>
<td>Oleic</td>
<td>18:1</td>
<td>62.5-65.3</td>
<td>11-43</td>
<td>43-49.8</td>
<td>25.2</td>
<td>43.1-45.79</td>
</tr>
<tr>
<td>Linoleic</td>
<td>18:2</td>
<td>19.2-22</td>
<td>44-74</td>
<td>11-12.26</td>
<td>55.4</td>
<td>32.27-34.4</td>
</tr>
<tr>
<td>Linolenic</td>
<td>18:3</td>
<td>8-9</td>
<td>0-0.8</td>
<td>0.1-0.54</td>
<td>5</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Table 4.1: Probability of different fatty acids in the triglyceride of the investigated vegetable oils [130].
Concluding from the varying yield, the present thesis mainly focuses on qualitative rather than a quantitative evaluation of the results. The required physical properties for \textit{SMD} calculation of the investigated crude vegetable oils of Table 4.1 at 21 °C along with those of other fuels are summarized in Table 4.2.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>(\sigma) [N/m]</th>
<th>(\mu \cdot 10^3) [Pa s]</th>
<th>(\rho) [kg/m(^3)]</th>
<th>(LHV) [MJ/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>0.025</td>
<td>2.5</td>
<td>835</td>
<td>43</td>
</tr>
<tr>
<td>Ethanol</td>
<td>0.0227</td>
<td>1.2</td>
<td>789</td>
<td>26.8</td>
</tr>
<tr>
<td>n-Heptane</td>
<td>0.0209</td>
<td>0.414</td>
<td>684</td>
<td>44.6</td>
</tr>
<tr>
<td>RME</td>
<td>0.0314</td>
<td>6.7</td>
<td>893</td>
<td>37.1</td>
</tr>
<tr>
<td>Jatropha</td>
<td>0.031</td>
<td>50</td>
<td>879</td>
<td>38</td>
</tr>
<tr>
<td>Palm</td>
<td>0.0332</td>
<td>106</td>
<td>890</td>
<td>37</td>
</tr>
<tr>
<td>Rapeeseed</td>
<td>0.0338</td>
<td>70</td>
<td>920</td>
<td>37</td>
</tr>
<tr>
<td>Soybean</td>
<td>0.0228</td>
<td>60</td>
<td>920</td>
<td>37.8</td>
</tr>
<tr>
<td>Sunflower</td>
<td>0.0227</td>
<td>49</td>
<td>919</td>
<td>37.8</td>
</tr>
</tbody>
</table>

Table 4.2: Relevant physical properties of the investigated fuels for \textit{SMD} calculation [130–136].

In practical devices, the combustion power demand is given rather than the fuel flow rate. Consequently, \textit{SMD} calculations were carried out keeping 15 kW of combustion power, which comes from the previous experimental studies [30] and is the target value in most of the investigated cases of the present thesis.

Figure 4.4 shows the \textit{SMD} characteristics at 21 °C at various atomizing pressures. Currently, ethanol is omitted, while it overlapped the characteristics of diesel oil. Based on the calculation, three groups can be clearly separated. Palm oil shows the highest \textit{SMD} at all atomizing pressures. Nevertheless, it was expected while palm oil is solid below 20–40 °C, depending on its origin. It is not a surprise that the other crude vegetable oils have similar atomization characteristics, while their viscosity and surface tension are also similar, shown previously in Tables 4.1 and 4.2. The fossil fuels along with the ethanol and the RME showed the most favorable atomization characteristics among the investigated fuels.

![Figure 4.4: SMD of different fuels at various atomizing pressures.](image)

Having a low \textit{SMD} does not necessarily mean a complete evaporation in the mixing tube. Volatility is also of importance which is usually in negative correlation with the
carbon chain length. The temperature of combustion air was 700 K.

Evaporation of diesel oil and n-heptane droplets were calculated by the extensive data by Lefebvre [13]. The physical properties of ethanol are also well-known, therefore, the required data is gathered from high-quality web databases [137–140]. The required temperature-dependent properties of other fuels are limited in the literature. Therefore, only rapeseed and soybean oil were investigated from evaporation point of view, using the following literature data [135,141–146]. These data are summarized in Appendix C.

Equations 2.21–2.28 were used for calculating the evaporation of the droplets. Table 4.3 contains the results at 700 °C and $p_g = 0.818$ (critical pressure ratio, hence maximum investigated value), 0.333 (mean value of the limits by atomizing air discharge velocity), and 0.053 bar (minimum investigated value). Hence, the $SMD$ was determined for the fuels. Besides these, the ratio of the heat-up period to the total time of evaporation ($t_{hu} + t_{st}$) was also calculated, highlighting the fact that more volatile fuels reach the steady-state period earlier.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>$p_g$ [bar]</th>
<th>$SMD$ [µm]</th>
<th>$t_{hu}$ [ms]</th>
<th>$t_{hu} + t_{st}$ [ms]</th>
<th>$t_{hu}/(t_{hu} + t_{st})$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>0.818</td>
<td>14.70</td>
<td>0.46</td>
<td>0.98</td>
<td>0.471</td>
</tr>
<tr>
<td></td>
<td>0.333</td>
<td>22.34</td>
<td>1.06</td>
<td>2.25</td>
<td>0.471</td>
</tr>
<tr>
<td></td>
<td>0.053</td>
<td>59.53</td>
<td>7.54</td>
<td>16.01</td>
<td>0.471</td>
</tr>
<tr>
<td>Ethanol</td>
<td>0.818</td>
<td>14.00</td>
<td>0.14</td>
<td>0.61</td>
<td>0.086</td>
</tr>
<tr>
<td></td>
<td>0.333</td>
<td>22.13</td>
<td>0.35</td>
<td>4.02</td>
<td>0.086</td>
</tr>
<tr>
<td></td>
<td>0.053</td>
<td>62.48</td>
<td>2.76</td>
<td>32.04</td>
<td>0.086</td>
</tr>
<tr>
<td>n-Heptane</td>
<td>0.818</td>
<td>10.68</td>
<td>0.12</td>
<td>0.62</td>
<td>0.191</td>
</tr>
<tr>
<td></td>
<td>0.333</td>
<td>16.67</td>
<td>0.29</td>
<td>1.50</td>
<td>0.191</td>
</tr>
<tr>
<td></td>
<td>0.053</td>
<td>46.11</td>
<td>2.20</td>
<td>11.48</td>
<td>0.191</td>
</tr>
<tr>
<td>Rapeseed</td>
<td>0.818</td>
<td>86.68</td>
<td>29.75</td>
<td>43.46</td>
<td>0.684</td>
</tr>
<tr>
<td></td>
<td>0.333</td>
<td>117.16</td>
<td>54.35</td>
<td>79.40</td>
<td>0.684</td>
</tr>
<tr>
<td></td>
<td>0.053</td>
<td>254.70</td>
<td>256.84</td>
<td>375.25</td>
<td>0.684</td>
</tr>
<tr>
<td>Soybean</td>
<td>0.818</td>
<td>75.58</td>
<td>19.86</td>
<td>32.68</td>
<td>0.608</td>
</tr>
<tr>
<td></td>
<td>0.333</td>
<td>102.43</td>
<td>36.47</td>
<td>60.02</td>
<td>0.608</td>
</tr>
<tr>
<td></td>
<td>0.053</td>
<td>224.07</td>
<td>174.52</td>
<td>287.22</td>
<td>0.608</td>
</tr>
</tbody>
</table>

Table 4.3: Atomization and evaporation characteristics at different gauge pressures and 700 K combustion air temperature.

Now a dimensionless number can be introduced which express the ratio of the residence time of the droplet inside a mixing tube to the total time requirement of its evaporation. While it is not named in the technical literature, it is introduced as evaporation number, $Ev$.

$$Ev = \frac{t_{res}}{t_{evap}} = \frac{t_{res}}{t_{hu} + t_{st}} \quad (4.2)$$

Regardless of the currently applied restrictions during the calculation, $Ev$ means a general relation between $t_{res}$ and $t_{evap}$. $t_{res}$ was calculated by Eq. 4.1 while $t_{evap}$ was derived by Eqs. 2.21–2.28. If $Ev$ exceeds one, it means that the droplet could evaporate completely before the flame front. However, This condition does not necessarily help in pollutant emission prediction [33]. Burning droplets ($Ev < 1$) usually increase both CO and NOX emissions [2,66]. If $Ev > 1$ and the flow is turbulent, the fuel vapor mixes well with the combustion air, resulting in a condition that approaches the emission levels of
premixed gas combustion [32]. Higher $E_v$ values allow better mixing and gaseous fuels are theoretically characterized by $E_v = \infty$.

![Figure 4.5: Ratio of residence and evaporation time for different fuels, $T_{s,0} = 21$ °C, $T_{s,st} = 700$ K.](image)

Figure 4.5 shows $E_v$ versus $p_g$ for diesel oil, ethanol, n-heptane, soybean oil, and rapeseed oil, using a logarithmic scale. Here, unity and higher values mean that a single fuel droplet completely evaporates in the 75.5 mm long mixing tube during its flight, assuming that the flame front is anchored to the burner lip.

Diesel oil and n-heptane fulfill this condition at all investigated atomizing pressures. But droplets of ethanol below $p_g = 0.11$ bar, soybean oil, and rapeseed oil are present at the burner lip. Therefore, utilization of crude vegetable oils at room temperature may result in incomplete combustion.

### 4.2 Fuel and combustion air preheating

There are two options to avoid the adverse effect of present droplets at the flame front. One is to increase the fuel temperature by preheating. Hence, viscosity and surface tension drops, shown in Table 4.4, and it results in smaller $SMD_s$.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>$T$ [°C]</th>
<th>21</th>
<th>40</th>
<th>70</th>
<th>100</th>
<th>140</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rapeseed</td>
<td>$\sigma$ [N/m]</td>
<td>0.0338</td>
<td>0.0321</td>
<td>0.0298</td>
<td>0.0276</td>
<td>0.0247</td>
</tr>
<tr>
<td></td>
<td>$\mu \cdot 10^3$ [Pa s]</td>
<td>70</td>
<td>31</td>
<td>12.8</td>
<td>6.83</td>
<td>3.75</td>
</tr>
<tr>
<td></td>
<td>$\rho$ [kg/m$^3$]</td>
<td>920</td>
<td>902.7</td>
<td>884.8</td>
<td>865.8</td>
<td>839.5</td>
</tr>
<tr>
<td>Soybean</td>
<td>$\sigma$ [N/m]</td>
<td>0.0338</td>
<td>0.0324</td>
<td>0.03</td>
<td>0.0277</td>
<td>0.0251</td>
</tr>
<tr>
<td></td>
<td>$\mu \cdot 10^3$ [Pa s]</td>
<td>60</td>
<td>28.5</td>
<td>12.1</td>
<td>6.699</td>
<td>3.75</td>
</tr>
<tr>
<td></td>
<td>$\rho$ [kg/m$^3$]</td>
<td>920</td>
<td>906.1</td>
<td>877.9</td>
<td>868.9</td>
<td>843</td>
</tr>
</tbody>
</table>

Table 4.4: Temperature-dependent properties of rapeseed and soybean oils [142,143].

Therefore, the evaporation time is also reduced, shown in Fig. 4.6. At lower temperatures, rapeseed oil is less volatile than soybean oil. By increasing the fuel temperature (hence $T_{s,0}$ increases), the gap between them continuously decreases but remains. At 140 °C, rapeseed and soybean oils catch up the diesel oil, which is shown here for comparison reasons.
Consequently, sufficient preheating can result in favorable combustion properties of such low-volatile vegetable oils. Note that the required physical properties of the rapeseed oil were given up to 140 °C [143]. However, it was preheated to 150 °C during the measurements in Chapters 5 and 6. The values of the temperature-dependent surface tension, viscosity, and density are listed below in Table.

The second method is to increase the combustion air inlet temperature, which can be achieved by, e.g., using an internal heat exchanger in gas turbines. Due to the typical operation parameters, the latter method is limited to small scale units.

Similarly to Fig. 4.6, there is also a gap between the trends of soybean and rapeseed oil in Fig. 4.7, which decreases by increasing the combustion air temperature. Interestingly, at 1200 K the trends overlap each other. The rapeseed oil outperforms the soybean oil at 2000 K. However, both of them are still far from the evaporation characteristics of diesel oil, unlike in the previous case. There was no local optimum condition for atomization in the investigated cases, shown in Figs. 4.5-4.7. Therefore, the evaporation is enhanced overall by the increasing atomizing pressure.

Regardless that, combustion air preheating seems to be a useful tool to increase the
ratio of residence and evaporation time, but fuel preheating in the range of the current analysis was found to work better. Furthermore, 2000 K lies 500 K beyond of the melting point of typical structural steels. Therefore, an advanced cooling system and more expensive materials are required in such environment. Nevertheless, spontaneous ignition and other violent phenomena may occur which harm combustion safety. In practice, the energy demand of fuel preheating is negligible compared to combustion air preheating to such a great extent. Fuel temperature of $\sim 140^\circ\text{C}$ is usually high for conventional polymer fuel pipes, but metal pipes can easily withstand it.

### 4.3 Summary of the chapter

Various fossil and renewable fuels were investigated in the present chapter from evaporation point of view at various atomizing pressures. Hence, $p_g$ affects the initial droplet sizes and the residence time of the spray in the mixing tube as well. The original question was that: low $p_g$, high SMD, and high residence time or high $p_g$, low SMD, and low residence time is better from evaporation point of view. The difference between the SMDs of conventional fuels (diesel oil, ethanol, n-heptane, and RME) and crude vegetable oils (jatropha, palm, rapeseed, soybean, and sunflower) started from the magnitude of two to four which increased further by increasing $p_g$.

To reveal the evaporation characteristics of the investigated fuels, the evaporation number, $Ev$, was introduced. It is defined as the ratio of residence time inside the mixing tube – assuming that the flow velocity is high enough to prevent combustion here – to evaporation time, using the $D^2$ law. It was expected that several factor difference arise between the crude vegetable oils and the diesel oil, which was the reference fuel here. In order to overcome this problem, preheating of the fuel and the combustion air were both investigated. Due to the limited temperature-dependent data, rapeseed and soybean oils were investigated further. The qualitative analysis revealed that preheated crude vegetable oils can catch up to diesel oil at high injection temperatures. Combustion air preheating is also a favorable solution, however, it is less effective compared to fuel preheating. These results were used in Chapter 5 for the appropriate measurement setup to achieve efficient rapeseed oil combustion in the LPP burner.
Chapter 5

Combustion in a LPP burner

The core of the thesis is combusting liquid fuels in an LPP burner that was shown earlier in Fig. 4.1. The present chapter starts with the introduction of the measurement system which is followed by the blowout stability analysis of the burner by utilizing rapeseed and diesel oils with special emphasis on the swirl number. Then Section 5.3 discusses the effect of different diffusers and a converging nozzle on blowout stability utilizing diesel oil. Based on the preliminary results, diffuser configurations with different half cone angles are examined. Pollutant emission analysis is presented in Section 5.4, investigating the burner with both rapeseed and diesel oils and comparing the results to the actual Hungarian emission decree. In order to enhance the stability limits set by swirl number on rapeseed oil combustion, the effect of a diffuser, confinement, and elevated combustion power were investigated.

5.1 The measurement system

Figure 5.1: The combustion test rig.
Figure 5.1 shows the atmospheric test rig, equipped with the burner shown in Fig. 4.1. Both the standard diesel oil (EN 590:2014) and the crude rapeseed oil have a separate fuel line for easy changeover purposes. The need for this is detailed in Section 5.2. Note that crude sunflower oil was also used, nevertheless, its stable operating limit was confined to a small region. Moreover, it does not show any chemiluminescence intensity peak which differs from diesel oil combustion, unlike rapeseed oil. Therefore, it is excluded from the present chapter. The spectroscopic and utilization details are discussed in Section 6.2.

Notable contaminations were not found inside the burner and its environment after experiments in the stable operating regime, shown in Section 5.2, utilizing diesel and rapeseed oils. However, sunflower oil combustion resulted in silicon-like deposits in the exhaust pipe, which is in agreement with the literature [7]. Therefore, it is required to carry out long time tests to ensure the deposit-free operation while the chemical composition of the fuel contains minimal information regarding contaminations. It is believed that fuels with a low number of unsaturated bonds result in a cleaner combustion.

Rapeseed oil arrives at the fuel mixer through an electric preheater, reaching 150 °C temperature in order to enhance its atomization. As the target combustion power was 15 kW, the mass flow rate of the diesel oil with 43 MJ/kg lower heating value was 0.35 g/s. Consequently, the rapeseed oil flow rate was 0.4 g/s, while it has 37 MJ/kg lower heating value. The fuel consumption was measured by taking a 30-second-average with a scale. The properties of both fuels are listed in Table 5.1. It can be concluded that the preheating of rapeseed oil to 150 °C temperature makes the atomizing characteristics very similar to that of the room temperature diesel oil due to the similar density, viscosity, and surface tension.

<table>
<thead>
<tr>
<th>Property</th>
<th>diesel oil</th>
<th>rapeseed oil</th>
</tr>
</thead>
<tbody>
<tr>
<td>LHV MJ/kg</td>
<td>43</td>
<td>37</td>
</tr>
<tr>
<td>$\dot{m}$ [g/s]</td>
<td>0.35</td>
<td>0.4</td>
</tr>
<tr>
<td>$T_0$ [°C]</td>
<td>20</td>
<td>150</td>
</tr>
<tr>
<td>$\rho$ [kg/m³]</td>
<td>830</td>
<td>833</td>
</tr>
<tr>
<td>$\mu \cdot 10^3$ [Pa s]</td>
<td>3.45</td>
<td>3.28</td>
</tr>
<tr>
<td>$\sigma$ [N/m]</td>
<td>0.028</td>
<td>0.024</td>
</tr>
<tr>
<td>Carbon [m%]</td>
<td>85.9</td>
<td>77.6</td>
</tr>
<tr>
<td>Hydrogen [m%]</td>
<td>13.5</td>
<td>11.6</td>
</tr>
<tr>
<td>Oxygen [m%]</td>
<td>0.6</td>
<td>10.8</td>
</tr>
</tbody>
</table>

Table 5.1: Fuel properties.

The atomizing air passes from the compressed air system of the laboratory through a pressure reducer and a rotameter to the airblast atomizer at $p_g = 0.3 - 2.3$ bar. Interestingly, the measurement of the thrust of the atomizing air correlated exceptionally well with the adiabatic expansion theory, shown in Fig. 4.3. As a result, the characterization of the atomizing jet can be carried out analytically. Note that the PDA, showed in Chapter 3, is not able to measure such high discharge velocities.

The combustion air, which is supplied by a frequency controlled fan, passes via another rotameter. Then the combustion air enters the electric preheater and leaves it at 400 °C temperature. Note that Yoon et al. used the same combustion air inlet temperature in their experiments [90]. However, the pressure drop of the combustion air system was found to be low, but the readings of both rotameters were corrected to ambient conditions. $\lambda$ was calculated as the sum of the measured flow rates of combustion and atomizing air.
to the stoichiometric air flow rate demand of the fuel, not considering the amount of entrained air from the atmosphere. Therefore, the primary \( \lambda \) is used in this thesis (i.e., which is present in the mixing tube). The stoichiometric air demand is calculated from the composition of the utilized fuels, listed in 5.1. The presence of oxygen in the diesel fuel comes from the renewable proportion according to the EN 590:2014 standard. Other measurements with the presented test rig can be found in the literature [30]. Note that the spectrometer is shown in Fig. 5.1, regardless that it is the subject of Chapter 6.

The flue gas was sampled with a Testo 350 portable emission analyzer directly from the stack. Each pollutant emission was corrected to 15% \( O_2 \) level as the burner was originally designed for gas turbine application. Note that the uncertainty of the measurement configuration is summarized in Appendix A.

5.2 Experimental stability analysis

Since there are notable differences in combustion properties of diesel and rapeseed oils (e.g., flash point, volatility, and heating value) [141, 147], operation problems cannot be solved in all cases by only preheating the fuel further. Therefore, a stability analysis is required to determine the safe operating regimes of the system running on such a fuel [65,87,93]. Hence, a flame blowout measurement was performed. If these limitations have to be extended further, there are different solutions available, which are configured for the particular system (see, e.g., [65,148,149]). In this case, the combustion air flow rate was increased until flame blowout was reached, similar to Hassan et al. [94]. Note that they faced with stability problems, which were related to the swirl number (Eq. B.3). Hence, it is emphasized in the current section.

It was concluded from the preliminary measurements that a successful ignition of rapeseed oil was impossible at startup. Therefore, diesel oil flame was ignited and fired first for about half a minute. J. Sallevelt et al. faced with similar problems in a micro gas turbine application [11]. In the present case, the changeover was only possible after the mixing tube was heated up to orange glow, being a condition which dramatically enhances the evaporation of rapeseed oil droplets through a very intense radiation. Thereupon, the flame root was anchored to the burner lip, resulting in a straight flame shape. During the present measurements, the atomizing pressure was set first, then the flow velocity of the combustible mixture was increased from 5 m/s until flame blowout. Interestingly, for diesel oil combustion, a well-defined blowout limit was found around 18.2 m/s mixture flow velocity, regardless of atomizing pressure. Consequently, this is the point where the flow velocity overcomes the turbulent burning velocity. The calculation of these velocities is discussed in Section 5.3.

In Fig. 5.2, the borders represent the stability and investigation limits in the case of diesel and rapeseed oil combustion. The blowout limitation mentioned above corresponds to \( \lambda = 1.36 - 1.46 \), depending on the mass flow rate of the atomizing air. Below \( p_g = 0.3 \) bar, the atomization became inadequate while \( p_g = 2.3 \) bar was the highest investigated atomizing pressure. The minimum \( \lambda \) corresponds to 5.23 kg/h combustion air flow rate in order to avoid burner meltdown by flashback.

Above 1.6 bar atomizing pressure, the rapeseed oil flame was blown out under a straight flame condition and same \( \lambda \) than the diesel oil flame. Hence, from stability point of view, they are equivalent fuels in this region, shown in Fig. 5.2. The explanation of this is the high velocity atomizing jet significantly weakens the swirl intensity. Consequently, the swirl number remains below the critical value. This was the reason why the investigation
stopped at $p_g = 2.3$ bar.

Below 0.48 bar atomizing pressure, the rapeseed oil flame was not stable. Among the dimensionless numbers of Table 5.2 at $p_g = 0.48$, only the $MFR$ was significantly less (6.82) than that of the diesel oil combustion case (8.89). It can be explained by the higher mass flow rate of rapeseed oil due to the difference in the lower heating value of the fuels. In order to ensure sufficient atomization, $MFR \leq 10$ was suggested by Prof. Sébastien Candel (personal communication, Budapest, September 3, 2015).

Howbeit, at $p_g = 1.6$ bar and below, the stable operating limits become continuously narrower, shown in Fig. 5.2b. The reason was the reaching of the critical swirl number at $S = 0.52$. Above that, a bistable region can be recognized, due to the broadband excitation by the turbulent oscillation of the flame, shown in Fig. 5.3. Here, both straight (flame is attached to the lip) and V-shaped (flame is detached from the lip and forms a V-shaped flame root above the lip) flames appear and a random transition occurs between them. At the beginning of it, the straight flame shows up for longer periods while this trend reverses as the end of the bistable region is approached. It is estimated as $S = 0.81$ for diesel oil flames, shown in Fig. 5.2a. By further increasing
the swirl number, a fully developed V-shaped flame characterizes the burner, shown in Fig. 2.5. These findings are in good agreement with the literature data [1].

Although the flame was able to enter the bistable region in case of rapeseed oil combustion, a violent, self-excited oscillation started with continuously growing amplitudes. It was followed by appearing droplets, then flame extinction within half a minute. The possible characterizing phenomena are the following: the flame opens to V-shape, and as a consequence, the radiative heat transfer to the droplets in the mixing tube drastically decreases. In this way, the droplet lifetime increases, and for this reason, the partially evaporated droplets leave the lip of the mixing tube. Consequently, the evaporation extract heat from the flame, further lowering the radiative heat transfer from the flame to the droplets. This self-excited phenomenon leads to flame extinction.

Table 5.2 shows the calculated dimensionless numbers and the $SMD$ by using Eq. 3.4, respective to the investigated atomizing pressures. Based on Fig. 2.2, it is evident that the corresponding initial liquid breakup mechanism was atomization in all cases considering the $Re_L$ and the $Oh_L$ numbers. The required $MFR$ for stable rapeseed oil combustion is probably slightly lower than 6.82. Therefore, the suggestion that $MFR \leq 10$ is required is confirmed here. Interestingly, the resulting $SMD$s of rapeseed oil are less than those of diesel oil atomization. However, its lower volatility, shown in Chapter 4, may result in less favorable mixing and higher pollutant emission. The latter one is discussed in Subsection 5.4.1. Additionally the corresponding FES measurement is presented in Section 6.4.

## 5.3 Flame stabilizing solutions

It was concluded above that the blowout bulk velocity of the burner in the case of diesel oil combustion was above 18.2 m/s. Those points were the last stable ones before the flame was blown out. The calculation method was not emphasized there, but it is detailed now: The volume flow rates of combustion and atomizing air were measured by using
two rotameters. To calculate the mass flow rates, the densities are required. While the atomizing pressure was measured and the temperature was found to be equal to that of the laboratory, its density can be calculated. The pressure increment of the combustion air caused by the fan was neglected. The sum of these mass flow rates equals to the total mass flow rate. In order to determine the average outflow velocity of the burner, the cross section of the mixing tube and the density are required. The former one can be measured, but the latter one requires further assumptions. In the case of a blowout, the flame is lifted up. Therefore, the ignition cannot start earlier, making the desired temperature of combustion air, 400 °C, a good guess at the mixing tube outlet. Regardless that the total mass flow rate is measured and free from assumptions, the mixing tube outlet velocity, \( w_{out} \), will be used instead due to its higher relevance in combustion engineering. It was derived as follows:

\[
\dot{m}_L = \frac{\dot{V}_{comb} \rho_0 + \dot{V}_{at} \left( \frac{p_g}{R_{sp} T_0} + \rho_g,673 \right) + \dot{m}_L}{A_{MT} \rho_{g,673}},
\]  

where \( \dot{V}_{comb} \) and \( \dot{V}_{at} \) are the volume flow rates of combustion and atomizing air, respectively. \( R_{sp} \) is the specific has constant, which is 287 J/kgK for air, \( \rho_{g,673} \) is the mass weighted mixture density at 673 K or 400 °C, \( A_{MT} \) is the cross section of the mixing tube, and \( \dot{m}_L \) is the mass flow rate of the liquid fuel. The measurement uncertainties are shown in the respective figures. Note that the present section is confined to the utilization of diesel oil only. The forthcoming section discusses the combustion and pollutant emission of both fuels.

The upcoming subsections introduce different diffusers and a converging nozzle fitted to the burner lip for flame stabilization. Regardless of their area ratio and averaged outlet velocity, the mixing tube outlet velocity is calculated in all cases for easier evaluation and comparison. The reason behind this is diffusers not necessarily have an attached flow in all cases, shown in Fig. 5.4.
5.3.1 Analysis of various burner configurations

The present subsection investigates the blowout stability of the original burner, two diffusers, and a converging nozzle attached to the burner lip. All these configurations are shown in Fig. 5.5.

The reason behind selecting different modifications was to measure the effect of various flow fields on the flame stability. The burner is slightly longer than the combustion air preheating chamber, shown in Fig. 5.5a. Hence, the flame surely detaches from the preheater. From fluid dynamics point of view, the application of a diffuser seems less attractive while the OZR already behaves like a virtual diffuser in the global flow field, shown in Fig. 2.5. Its role is to expand the flow. Hence, the diffusers are only the substitutions of the ORZ with a solid material if the flow is inviscid [78]. Therefore, the most straightforward geometry and angle was chosen in Fig. 5.5b. Figure 5.5c presents a diffuser variation with smaller half cone angle. Here, there is an annular cavity next to
the burner which hosts a recirculation zone. Hence, the flame can attach to the diffuser wall easier. The last configuration was achieved by turning the diffuser upside down.

Figure 5.6 shows the preliminary blowout stability investigation. However, the bistable region was also investigated here, but it is not presented while Subsection 5.3.2 shows a more detailed, systematic measurement series. The aim of this subsection is to understand the role of the diffusers in combustion stability.

![Blowout stability of the investigated configurations.](figure)

The axes caption choice requires some explanation. It is known that the mixing tube outlet velocity contains the mass flow rate of the atomizing air. Hence, they are not necessarily independent variables. If the atomizing free jet transfers its excess kinetic energy to the combustion air inside the mixing tube, the origin of air does not affect the combustion itself. While the length of the mixing tube is 75.5 mm and the hydraulic diameter of the atomizing annulus-shaped free jet is 0.6 mm, the dimensionless axial distance is more than 125. Therefore, it is a good assumption that the atomizing pressure affects the flow field near the nozzle, and its effect is faded to the bulk flow until it reaches the burner lip. Consequently, they can be handled as independent variables.

Fig. 5.6 shows the blowout stability (i.e., the mixing tube exit velocity where the combustion was yet stable) of various burner configurations. In this measurement series, only the 45° diffuser was outperformed slightly by the original burner below $p_g = 0.75$ bar. The 17° diffuser, incorporating a cavity next to the burner lip, showed great stability limits compared to the previous two configurations. The difference between the three setups fades above $p_g = 1.6$ bar while the flame remains straight before the blowout. It can be explained by the low swirl numbers, discussed in Section 5.2. Surprisingly, the converging nozzle was superior to all other configurations. Figure 5.7 shows the corresponding flame near the blowout limit.

It was mentioned in Subsection 2.4.2 that the flame stabilization is enhanced by the IRZ in the case of LPP burners. Nevertheless, a recirculation zone which transfers heat back to the upstream mixture does not need to be placed necessarily outside of the burner. This converging nozzle configuration incorporates a cavity. Burners of this type are known in the literature as trapped vortex combustors or TVCs in short [150]. Note that the term combustor is the shortened variant of the combustion chamber. The flame of Fig. 5.7 shows an annular-shaped bright area near the wall which indicates the stabilizing vortex which anchors the flame. Apparently, the flame shape remained straight in all these cases; no bistable and V-shaped regime were possible due to the converging outlet.
This outstandingly high stability characteristic also has a drawback. The addition of the last 3-4 kg/h combustion air, or equivalently the last 2.85-3.8 m/s mixing tube outlet velocity, before blowout resulted in incomplete combustion of the fuel and a spectacular blue smoke was generated due to the higher flow rates. If the emission of unburnt hydrocarbons and the efficiency are of paramount importance, it is recommended to use the diffusers, detailed in the forthcoming subsection. If the stability comes first, like in military applications, the TVC seems to be the superior solution.

Based on Fig. 5.6, it is clear that the application of diffusers has a significant impact on flame stabilization. Therefore, videos at 30 frames per second were recorded to gather more information about the flame blowout mechanism at various atomizing pressures and burner configurations, shown in Figs. 5.8 and 5.9.

The first frame in Fig. 5.8 shows the flame 1 second before the blowout while the
remaining ones are the last frames before it. It can be seen that the flame detaches from the burner lip, shown in the second frame and attaches back later. Frame 6 shows the effect of reduced heat transfer between the IRZ and the upstream mixture as the flame lifts up. The lack of sufficiently preheated upstream mixture leads to a blowout. While the whole blowout process took less than 2/30 second, high-speed videos at 480 frames per second were also recorded. The results are shown in Appendix D.

Figure 5.9 shows the blowout phenomenon in the case of an applied 45° diffuser. The first frame was also taken 1 s before the blowout, similar to the previous case, to show a relatively stable flame. The stabilizer resulted in a V-shaped flame, which was present until the last frames; significant distortion was not identified. Only the oscillation of IRZ is notable. The last frame is of primary interest while it reveals the corresponding blowout mechanism. By comparing it to the last frame of Fig. 5.8, it can be seen that combustion is present near the diffuser wall before blowout.

Figure 5.9: Sequences from blowout with a 45° diffuser, $p_g = 0.7$ bar.

It can be explained by the fact that the boundary layer provides lower flow velocities than it is present in the main flow, hence, the burning velocity was not overcame by the flow velocity. Therefore, this region could relight the upstream mixture and made it clear why the application of a diffuser can maintain higher blowout limits. Now it makes sense that the substitution of the ORZ with solid material may significantly affect the combustion performance. The shape of the ORZ is determined by the actual flow conditions; it can distort and even shrink. Howbeit, if a diffuser is applied, its virtual shape is fixed, resulting in an additional design parameter. Now we have arrived at the forthcoming subsection, namely, finding the best angle for the diffuser.

5.3.2 Application of diffusers

Diffusers of 16 mm slant height and 0°, 15°, 30°, 45°, and 60° half cone angle were investigated in this subsection, shown in Fig. 5.10. While all of them were milled from a cylinder of the same size, the added material may cool the flame through its increased heat capacity. It was a partial reason why a 0° variation was involved; nevertheless, it only provides an additional 33% length for the mixing tube.

It was concluded in Section 5.2 that above $p_g = 1.6$ bar the bistable region is not achieved. It was also supported by Fig. 5.6 in the preliminary measurements. Therefore,
the range of $p_g = 0.3 - 1.6$ was investigated. Figure 5.11 shows the result of blowout stability analysis.

![Figure 5.11: Blowout stability of the diffusers.](image)

The slight difference between the atomizing pressures around a target value arise from the fact that the atomizing air volume flow rate was set by a rotameter. The data recorded by the pressure transducer were evaluated later. The addition of the $0^\circ$ 'diffuser' did not influence the blowout characteristic of the burner considerably. Therefore, the mixing tube length was proven to be appropriately designed. The $60^\circ$ diffuser was found to be too steep to enhance flame stability significantly while the flame was not able to attach to the diffuser wall. Nevertheless, the $45^\circ$ diffuser was superior to the mentioned three configurations, showing a continuous enhancement in the investigated range. Nevertheless, $15^\circ$ and $30^\circ$ diffusers outperformed all the other variations. There is no significant difference between them and both configuration resulted in a 42% improvement on average. Due to the turbulent nature of the flame and the varying load in practical combustion chambers, it is expected that there is no half cone angle between $15^\circ$ and $30^\circ$ which provides even superior performance under all conditions. However, this result is highly valuable for combustion engineers.

The detection of the start of the bistable region is facilitated by the formation of the IRZ. While the pressure inside it is low, the float of Rotamer 1 in Fig. 5.1 jumps and the total mass flow rate increases. Apparently, the shift between the two flame shapes also results in an oscillation of the mass flow rate, hence, the float. Figure 5.12 shows the calculated mixing tube outlet velocities for the different diffusers.

Interestingly, all the trends go together, indicating that the beginning of the bistable
zone is not affected by the presence of the diffusers. It leads to the conclusion that the bistable region depends only on the swirl number. The limited data shown here is due to the flame was blown out before it reached the bistable region in certain cases. Note that such pressure fluctuations are not favorable in industrial combustion chambers while it may damage them or at least shortens the interval between overhauls. This bistable region is handled similarly to critical speed in rotordynamics. It is desired to pass through it as fast as possible.

Next, the start of the V-shaped region is shown in Fig. 5.13. As a consequence of the previous paragraph, the V-shaped flame was also not reached by flames which were blew out earlier than they entered the bistable region.

Similar to Fig. 5.12, the tendencies show no sensitivity for the half cone angle of the diffusers. Here, a progressive characteristics can be seen as the increasing atomizing pressure lowers $S$. This measurement result is also supported by the calculated swirl number characteristics, shown in Fig. 5.2.

Figure 5.14 compares the stability limits by the original and the 15° diffuser. It shows more exactly why the original burner was not able to provide a V-shaped flame at all atomizing pressures.

It was shown in Figs. 5.12 and 5.13 that the bistable region is primarily determined by the swirl number. Due to the limitation set by blowout, the V-shaped flame was observed
until $p_g = 0.45$ bar and the bistable region was limited to $p_g = 0.81$ bar in the original case. These trends can be followed in the whole investigated range in the case of the $15^\circ$ diffuser while its blowout characteristic was significantly higher. Figure 5.14 reveals that the presented sequences in Fig. 5.8 correspond to a bistable region. This is the reason why the first frame in Fig. 5.8 showed a straight and the last frames V-shaped flame.

Seeing the outstanding performance of the diffusers, rapeseed oil combustion was tried to exceed the critical swirl number by using them. The result was a failure which was already suggested by Figs. 5.12 and 5.13. However, this goal is further investigated in Subsection 5.4.2.

## 5.4 Pollutant emission analysis

According to Nakamura et al. [33], short droplet lifetime and complete evaporation do not mean clean combustion necessarily. Therefore, pollutant emission analysis is required to check the combustion chamber whether it is in line with the standards or not. Subsection 5.4.1 compares the pollutant emission of diesel and rapeseed oil combustion without diffusers. The respective flame stability analysis was already shown in Section 5.2. The next subsection incorporates the investigation of three modifications in the burner setup. Namely, the effect of applying a $45^\circ$ diffuser, confining the flame, and elevating the combustion power.

### 5.4.1 Comparison of rapeseed oil and diesel oil combustion

Figures 5.15 and 5.16 summarized the results of the flue gas analysis which was carried out in parallel with stability and FES measurement, discussed in Sections 5.2 and 6.4, respectively. Therefore, certain phenomena will be explained by flame stability.

The formation of NO\textsubscript{X} and CO is affected by atomization, mixing, volume and temperature-distribution of the reaction zone, and the residence time. During the measurements, atomizing pressure and $\lambda$ were varied which influence the emission levels through these phenomena.

In the case of diesel oil, the most complete combustion was located at $\lambda = 0.75$ and $p_g = 1.12$ bar. The CO emission was the lowest and NO\textsubscript{X} emission was nearly the highest
here, shown in Fig. 5.15. Starting from this measurement point the CO emission increases, while the NO\textsubscript{X} emission decreases by increasing \( p_g \). Here, the increasing thrust of the atomizing jet, shown in Table 5.2, now obstruct the mixing of the evaporated droplets with the combustion air. The higher velocity near the axis results in a less complete combustion and less residence time, enhancing the formation of CO and reducing the NO\textsubscript{X} emission. The average droplet size increases by decreasing \( p_g \). Therefore, the spray requires more time for complete evaporation, resulting in incomplete prevaporization. Hence, droplets appear in the flame and the combusting mixture becomes heterogeneous due to the lack of mixing, increasing the CO emission. In this way, the flame becomes more luminous. Due to the evaporating droplets, the flame temperature decreases and leads to lower NO\textsubscript{X} emission. The CO emission increases with \( \lambda \), regardless of \( p_g \). Increased combustion air flow rates also increase \( S \) until the flame reaches V shape. The higher flame volume is characterized by lower temperatures and higher bulk velocity through the increased combustion air flow rate. Hence, the combustion becomes less complete, significantly increasing CO emission. The NO\textsubscript{X} concentration drastically reduces in parallel with the CO. Note that the blowout stability limit is near under V-shaped conditions.
Rapeseed oil combustion was not possible under all the investigated $p_g - \lambda$ conditions, as it was discussed in Section 5.2. Based on Fig. 5.16, there is no precursor in the CO or NO$_X$ plots of exceeding the critical swirl number and lose flame stability. The most complete combustion was achieved at $\lambda = 0.75$ and $p_g = 0.9$ bar which is close to that of diesel oil combustion.

Figure 5.16: a) NO$_X$ and b) CO emission of rapeseed oil combustion.

The change of pollutant emissions was found to be similar in the case of both fuels. The only exception is the sudden NO$_X$ increase at low $p_g$ and $\lambda$. The reason behind this is the low volatility of the rapeseed oil in combination with the higher required residence time for evaporation. The resulting inhomogeneous mixture leads to the formation of hot spots, significantly facilitating NO$_X$ formation [66]. The combustion of the more volatile diesel oil showed decreasing NO$_X$ emission in this region. Note that the CO emission of rapeseed oil combustion increases to a greater extent with $p_g$ than that of diesel oil combustion. The explanation of this phenomenon is the following: at high atomizing pressures, the heat capacity of the atomizing flow increases, and the expansion of the atomizing jet ends at lower temperatures. Hence, the atomizing jet cools the preheated rapeseed oil droplets, encumbering secondary atomization and evaporation through the increased viscosity and
surface tension. Furthermore, the velocity of the small droplets is high, resulting in a low residence time which facilitates the CO formation. This phenomenon is also present in the case of diesel oil combustion, but to a lower extent while its volatility is less sensitive to temperature changes.

In order to objectively compare the overall emission levels, the current Hungarian decree for gas turbines of >140 kW combustion power is used (7/1999 (VII. 21.) KöM Decree). It allows 200 mg/m$^3$ NO$_X$ and 100 mg/m$^3$ CO concentration at 15% O$_2$. The emissions of Figs. 5.15 and 5.16 were compared to the limitations, using the following formula:

$$EM = \frac{EM_{NOx}/EM_{NOx,limit} + EM_{CO}/EM_{CO,limit}}{2},$$  \hspace{1cm} (5.2)

where $EM$ denotes the averaged emission and the subscripts refer to the specific pollutants and the limitations. Note that all the pollutant emission values were converted from ppm to mg/m$^3$. The results are shown in Fig. 5.17.

Figure 5.17: Overall emission of a) diesel oil and b) rapeseed oil combustion.

It can be concluded that CO emission governs the overall emissions in the present
configuration. The lowest emission is located at $p_g = 1.12$ and $\lambda = 0.75$ for both fuels, and the trends are also similar. The only notable difference is the cooling of the rapeseed oil droplets due to the low temperature atomizing jet, as it was mentioned above. This result indicates that preheating temperature of 150 °C makes the rapeseed oil similar to diesel oil from combustion point of view in the stable operating region.

5.4.2 Effect of the diffuser, the confinement, and the elevated combustion power on pollutant emissions

It was concluded in Subsection 5.3.2 that flame stability can be enhanced by applying diffusers in the case of diesel oil combustion. Nevertheless, pollutant emissions are of primary importance in the civil sector. Therefore, the effect of the 45° diffuser was investigated first at $\lambda = 1.15$ and 1.35. Diesel oil was utilized in Figs. 5.18 and 5.19 while the rest of this subsection compares diesel and rapeseed oil combustion under various conditions.

![Figure 5.18: Straight (left) and V-shaped (middle) flame of the original burner and the V-shaped flame with the 45° diffuser (right).](image)

![Figure 5.19: Emissions of the original burner and with a 45° diffuser.](image)

The corresponding flame shapes are straight and fully developed V, respectively, to avoid the bistable region, discussed in Subsection 5.3.2. It is expected that the applied
diffuser have a negligible impact on pollutant emissions while the flow is detached from that. The corresponding images are shown in Fig. 5.18. The V-shaped flame was flattened and became less luminous with the 45° diffuser.

According to the flue gas analysis at $\lambda = 1.15$, the applied diffuser resulted in slightly higher NO$_X$ and lower CO emission compared to that of the original configuration, shown in Fig. 5.19. However, the differences at V-shaped flames were more significant.

By increasing $\lambda$, the NO$_X$ emission of the V-shaped flame falls drastically, while the CO emission doubles. The explanation of this phenomenon is the entrained ambient air at the root of the flame. The diffuser prevents the entrainment, therefore, the NO$_X$ and CO emissions remained similar to the 1.15 air-fuel equivalence ratio case. The slight decrease arises from the leaner operation, which effect was shown earlier in Fig. 2.9. Therefore, a diffuser is also practical if the burner operates in a relatively cold environment to avoid the excessive CO formation.

![Figure 5.20: The burner lip with the confining tube.](image)

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>diesel oil</td>
<td>15 kW</td>
</tr>
<tr>
<td>diesel oil</td>
<td>16.8 kW</td>
</tr>
<tr>
<td>rapeseed oil</td>
<td>15 kW</td>
</tr>
<tr>
<td>rapeseed oil</td>
<td>16.8 kW</td>
</tr>
</tbody>
</table>

Table 5.3: Combustion of diesel oil and rapeseed oil with and without confinement and at 15 and 16.8 kW.
Next, the investigation of confinement and elevated combustion power is discussed. As for the previous configuration, a piece of steel pipe was placed on the burner to prevent the cold air entrainment at the flame root, shown in Fig. 5.20. The motivation behind these ideas was the investigation of their effects on the stability of rapeseed oil combustion. Therefore, $\lambda = 1$ and $p_g = 0.9$ bar were chosen as operating parameters, while these are close to the extinction limit of rapeseed oil combustion through the appearing bistable zone. Table 5.3 contains the corresponding images.

The value of 16.8 kW was calculated based on Fig. 5.2 to provide a condition where an appearing bistable flame is expected. To ensure that the mentioned stability limit of rapeseed oil combustion is determined by only the swirl number, $\lambda$ was kept at constant. Firstly, the combustion power was elevated, then $\lambda$ was adjusted to the desired condition by increasing the combustion air flow rate. The diesel oil flame remained stable, as it was predicted by Fig. 5.2. Nevertheless, rapeseed oil flame becomes unstable and blew out similarly as in the previous case. The elevated combustion power, hence, the increased radiative energy which evaporated the droplets, was insufficient to stabilize the flame under these conditions, shown in the top right image in Table 5.3.

![Figure 5.21: Measured pollutant emission at the stack at 15% O$_2$ level.](image)

By using the finding of Fig. 5.19, a confining tube was applied which prevents the entrainment of ambient air at the flame front. These measurements ended with the same conclusion as the previous one. Confinement does not help the situation while a bistable flame appeared which automatically results in a flame blowout. As a conclusion, stable combustion of rapeseed oil requires swirl numbers below the critical value which ensures a straight flame. The corresponding pollutant emissions are measured, shown in Fig. 5.21.

Confinement ensures lower CO and higher NO$_X$ emissions in each case due to the exclusion of ambient air. At 15 kW combustion power, the emission levels remained relatively similar. An exception is the unconfined diesel oil combustion which resulted in a lower NO$_X$ and a higher CO emission.

At 16.8 kW, all configurations showed similar pollutant emissions, except the unconfined rapeseed oil flame. It is noteworthy that the gas analyzers were able to detect the unstable operation through the excessive CO content but only a few seconds before the blowout. However, the confined and also unstable flame remained undetected by the gas analyzers. Now it is shown in detail that an effective flame control requires low-response measuring devices, as it was proposed at the end of the previous subsection.
5.5 Summary of the chapter

Firstly, the atmospheric combustion test rig was introduced in this chapter which provided a solid basis to investigate the effect of various modifications on combustion stability. The two utilized fuels were diesel and rapeseed oils. Section 5.2 showed that the blowout stability characteristics of both fuels are similar above $p_g = 1.6$ bar. Below that, another limiting factor appeared in the case of rapeseed oil combustion. This flame could not enter the bistable region and reach a fully developed V-shaped flame shape. The altering flame shape from straight to V lead to less heat radiated back that enhanced the droplet vaporization. Appearing droplets at the flame front cooled it down which resulted in a flame blowout through this self-excited phenomenon. Based on the finding of Section 5.2 and Subsection 5.4.2, it can be concluded that stable combustion of rapeseed oil was limited by the critical swirl number. Another limitation was the insufficient atomization which can be found below $p_g = 0.48$ bar.

Secondly, the analysis of various diffuser configurations was performed. The bistable region was found to be governed by only the swirl number and thus the diffuser half cone angle did not affect it. But the situation is different in the case of blowout stability. Among all diffusers, $15^\circ$ and $30^\circ$ performed the best, improving the blowout stability by 42% on average in both cases.

Thirdly, a pollutant emission analysis was carried out by utilizing rapeseed and diesel oils. Note that this measurement was carried out in parallel with the above-mentioned stability analysis and FES investigation, discussed in Section 6.4. By considering the 7/1999 KöM Decree on emissions of gas turbines, the averaged NO$_X$, and CO showed similar trends. The lowest emission was located at $p_g = 1.12$ bar and $\lambda = 0.75$ for both fuels. It turned out that CO governs the averaged pollutant emissions. Hence, rapeseed oil combustion was less favorable at high atomizing pressures due to the adverse effect of the atomizing jet on the secondary droplet atomization and evaporation. Regardless that low $p_g$ and low $\lambda$ showed increased NOX emission due to the heterogeneous combusted mixture. The averaged emission hides this information while this pollutant was already low due to the entrained cold air.

Neither a diffuser, not confinement, nor elevated combustion power was able to improve the flame stability of rapeseed oil combustion. Therefore, the superior role of the swirl number on combustion stability was found.

Figures 5.8 and 5.9 compared the blowout mechanism of the original burner with the configuration where a subsequent $45^\circ$ diffuser was used. By combining this result with the blowout stability characteristics of diffusers with various half cone angle, shown in Fig. 5.11, the following thesis can be stated:

2. Thesis

Blowout stability of V-shaped flames characterized by strong swirl in lean premixed prevaporized burners enhances by applying a subsequent diffuser on the burner lip, which substitutes the outer recirculation zone. In this case, the appearing boundary layer near its wall ensures the improvement of the blowout stability.
Chapter 6

Spectrometry

In Section 2.5, various approaches were introduced for flame diagnostics, emphasizing the importance of real-time monitoring of the local $\lambda$. Firstly, the present chapter details the properties of the applied spectrometer. Secondly, it introduces the evaluation methods, detailing various corrections of the spectra. Thirdly, the spatial FES measurement is shown for three operating conditions to locate the best measurement position through the plots of OH*, CH*, and C2,516 and the ratios of them along the axis.Fourthly, crude rapeseed oil combustion is analyzed and compared with diesel oil under various equivalence ratios and atomizing pressures.

6.1 The spectrometer

A fixed spectrometer was used for the spectroscopic analysis of the investigated flames, which was manufactured by OpLab Ltd., shown in Fig. 6.1. The focal length of the 20 mm diameter quartz objective is 0.5 m which means a line of sight measurement with 5 mm diameter at the focus, shown schematically in Fig. 5.1. The light sensor of the spectrometer is an n-channel metal-oxide semiconductor field-effect transistor (nMOS). The diffraction grid determines the range of the spectrum, which was 260 – 580 nm (UV–VIS, hence, the resolution is 0.3125 nm).

The Hamatsu nMOS S3904-1024Q detector (here, the 1024 denotes the number of light sensitive pixels on the sensor) has a non-uniform characteristic in the mentioned
spectral region. Therefore, correction of spectra was performed based on its factory data sheet besides the dark current correction. The integration time of the spectrometer is adjustable in its software, which was set to 100 ms. In each measurement case, 64 measured spectra were averaged by intensity. Further details of this device can be found in the literature [151].

6.2 Evaluation methods

It was mentioned in Subsection 2.5.1 that the literature contains no information regarding chemiluminescence emission investigation of crude vegetable oils. However, these fuels along with the diesel oil are considered a long-chain hydrocarbons. Since the present thesis investigates steady-state operation at chemically high temperatures (>1000 K), one expects that the spectra of these flames should be similar. The measuring volume of the spectrometer was focused to the axis, slightly above the burner lip.

The combustion of standard diesel fuel resulted in typical chemiluminescence emission spectra, showing the OH$^*$, CH$^*$, and C$_{2,516}^*$ chemiluminescence intensity peaks of the flame clearly. Similar characteristics were observed during rapeseed oil combustion, shown in Fig. 6.2. However, at 554 nm, a new potential peak was identified for further analysis, which is clearly observable in all the rapeseed oil combustion cases. Besides showing scaled trends in the spectral neighborhood compared to that of diesel oil combustion, this peak was outstanding probably due to the higher concentration of specifically excited C$_2^*$. It fits to the A$^3\Pi_g$–X$^3\Pi_u$ transition of S(2,3) of the Swan system, and no other possible radical candidate was found at this wavelength [152]. Consequently, the newly identified peak is denoted as C$_{2,554}^*$ in the followings. According to the best knowledge of the author, there is no sign of using this chemiluminescence peak in the literature of FES. These results were discussed with reaction kinetics experts Tamás Varga and prof. Tamás Turányi from Eötvös Loránd University Budapest. They concluded that this peak was not expected under these conditions and it requires more investigation from both experimental and reaction simulation side (personal communication, Budapest, April, 2014).

![Figure 6.2: Spectra of rapeseed oil and diesel oil combustion at $p_g = 1$bar and $\lambda = 1.15$. Note that only dark current correction was performed for better visibility.](image)

The chemiluminescence intensity of OH$^*$, CH$^*$, C$_{2,516}^*$, and C$_{2,554}^*$ were measured at 309, 430, 516, and 554 nm, respectively. Interestingly, the chemiluminescence intensity at C$_{2,554}^*$ is unusual in flame diagnostics. Further explanations of the latter choice are
described in Section 6.4. It is known that the OH$^*$ signal can be measured directly as it is free from superposition of chemiluminescence intensity of other radicals and the black body radiation of soot particles at practical combustion temperatures. This is not true for the other radicals mentioned because of the known broadband emission of CO$_2^*$ in the range of 310-600 nm [109]. Furthermore, the black body radiation spectrum should be taken into account, which intensifies with the flame luminosity, shown in Fig. 6.3. At $p_g = 0.35$ and 1 bar only the intensities change in a consequent manner. A possible explanation for this phenomenon is that the finer droplets evaporate faster. Thus, the combustible mixture was ignited earlier in the mixing tube.

The typical strategy of chemiluminescence intensity correction is to simply subtract a value near the peak, using either the rest of the measured spectrum or a combination of bandpass filters [109, 110, 117]. These should be selected carefully in order to make sure that there will be no chemiluminescence peak present in the spectrum at the selected wavelengths under any possible conditions. Therefore, this mentioned strategy was slightly updated, because the black body radiation spectrum is becoming more significant at rich conditions, shown in Fig. 6.3. It distorts the correction for CH$^*$, C$_2^*$, and C$_2^*$ as well. Hence, two 5 nm wide bands were selected in both higher and lower wavelengths for each peak.

![Figure 6.3: Effect of $\lambda$ and $p_g$ on the chemiluminescence emission spectra through the appearance of soot.](image)

<table>
<thead>
<tr>
<th>Emitters</th>
<th>Peak wavelength</th>
<th>Lower correction</th>
<th>Higher correction</th>
</tr>
</thead>
<tbody>
<tr>
<td>OH$^*$</td>
<td>309</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>CH$^*$</td>
<td>431.4</td>
<td>408-413</td>
<td>445-450</td>
</tr>
<tr>
<td>C$_2^*$</td>
<td>516.5</td>
<td>489-494</td>
<td>523-528</td>
</tr>
<tr>
<td>C$_2^*$</td>
<td>554.1</td>
<td>529-534</td>
<td>570-575</td>
</tr>
</tbody>
</table>

Table 6.1: List of the considered chemiluminescence emitters and their respective higher and lower 5 nm wide bands for correction.

These regions were first selected based on the raw spectra, then they were fine-tuned to avoid superposition with weaker emitters of organic flames, using a comprehensive spectroscopic table [152]. These selected bands are summarized in Table 6.1. The averaging was performed at first, then proper weighting of the bands were carried out in order to
estimate the background emission at the wavelength of the selected peaks. The described method is recommended first by the author in the field of FES analysis. While Section 6.3 analyzes spatial distribution of OH*, CH*, and C2,516 at \( \lambda = 1.15 \) utilizing diesel oil, the newly developed method becomes more significant in Section 6.4, where \( \lambda \) varies from 0.7 to 1.5, involving combustion regimes from rich to lean.

Despite that crude sunflower oil combustion may contaminate the burner [7], a preliminary measurement was carried out with this fuel as well. Interestingly, stable flame was hardly achieved, just under conditions that belong to the limitations of the combustion test rig (i.e., \( \lambda = 0.9 - 1.1 \), \( p_g = 3 \) bar). Therefore, a spectrum was recorded and compared with diesel oil combustion at 3 bar atomizing pressure and \( \lambda = 1.15 \), shown in Fig. 6.4. It is clear that the captured spectra are very close to each other. Hence, no further analysis was carried out utilizing crude sunflower oil due to its poor combustion properties and the lack of differences in the spectrum like the C2,554 in the case of rapeseed oil combustion.

![Spectra of crude sunflower oil and diesel oil combustion at \( p_g = 3 \) bar and \( \lambda = 1.15 \).](image)

**Figure 6.4:** Spectra of crude sunflower oil and diesel oil combustion at \( p_g = 3 \) bar and \( \lambda = 1.15 \).

### 6.3 Spatial FES measurement

Chemiluminescence intensity was measured horizontally at thirteen points and nineteen points vertically with 5 mm spacing in both directions by moving the test rig, shown in Fig. 5.1. The position of the spectrometer remained fixed while the rig was moved. The estimated accuracy of this system is 1 mm in both directions. The atomizing pressures were 0.35, 0.7, and 1 bar. For post-processing, linear interpolation was used on the contour plots to show a better visual representation. Each plot was halved vertically and joined together with a halved photograph to improve the understanding and interpretation of the observed flame structure, as shown in Fig. 6.5. The goal of this analysis was to compare the effect of varying atomizing pressure on spectral structure of the flame at \( \lambda = 1.15 \). If the fuel vapor and the combustion air mix well, the result is not expected to depend on \( p_g \).

The intensity range was set from zero to 700 to produce easily-comparable plots. At 0.35 bar atomizing pressure, the maximum CH* intensity was higher than 700, so it was set to 1300. Here, the background noise correction, arising mainly from CO2*, was neglected while its effect was negligible based on the curve fitting method, discussed by Guyot et al. in ref. [110]. Obviously, the dark current correction was performed. Nevertheless, it is clear
that the chemiluminescence intensity of OH\(^*\), CH\(^*\), and C\(_{2,516}\) radicals, and the volume of intense combustion decrease with increasing atomizing pressure. The flame luminosity follows the same trend. Therefore, the chemiluminescence intensity gradients behave in the opposite way. The explanation of these phenomena is the decreasing swirl number by the increasing \(p_g\). At \(\lambda = 1.15\), there is enough oxygen available for combustion. The small droplets evaporate faster and the intensifying shear between the atomizing jet and the combustion air enhances the mixing of the fuel vapor which leads to the bluish color of the flame. Because of the larger droplet sizes at lower atomizing pressure result in slower evaporation and mixing, the less homogeneous mixture leads to an enhanced soot formation. It is indicated by the higher flame luminosity.

Chemiluminescence intensity ratios in such plots are not shown because the intensities drastically decreased in horizontal direction, resulting in extremely high uncertainties. This phenomenon was expected due to the large gradients in the temperature and the reaction rates in this direction. Vertically, the chemiluminescence intensities decreased in a remarkably lower extent. However, at 32.5 mm axial distance, the uncertainty became 100% at 95% level of significance, and it increased further. Consequently, the investigated region is limited to the range of 2.5-27.5 mm downstream from the burner lip.
In the upcoming part, the best position for the spectrometer is discussed. There is no doubt that it should point at the axis in the case of an axisymmetrical burner. As for fluid fueled burners, the downstream distance of the control volume from the burner lip is not evident. The centrally injected droplets can be present near the axis even after the lip, hence, quenching the flame [113]. Figure 6.6 presents the corrected chemiluminescences of \( \text{OH}^*/\text{CH}^* \), \( \text{OH}^*/\text{C}_2\text{H}_{5,16}^* \), and \( \text{CH}^*/\text{C}_2\text{H}_{5,16}^* \). As for \( \text{CH}^* \) and \( \text{C}_2\text{H}_{5,16}^* \) correction, intensity values at 450 nm and 528 nm were simply subtracted, similarly to the work of Docquier et al. [117]. The error bars are respective to 95% level of significance.

Figure 6.6: Corrected \( \text{OH}^*/\text{CH}^* \), \( \text{OH}^*/\text{C}_2\text{H}_{5,16}^* \), \( \text{CH}^*/\text{C}_2\text{H}_{5,16}^* \) chemiluminescence ratios along the axis.

In Figure 6.6, the \( \text{OH}^*/\text{CH}^* \) and \( \text{OH}^*/\text{C}_2\text{H}_{5,16}^* \) chemiluminescence intensity ratios increase and become even more divergent with increasing axial distances. The previous one shows relatively similar chemiluminescence ratio of \( \text{OH}^* \) and \( \text{CH}^* \) just above the lip while the latter shows the closest ratios at 12.5 mm above the lip. The uncertainty at 27.5 mm axial distance remains below 20% in each case, and it decreases to 4.5% at the flame root quadratically. The \( \text{CH}^*/\text{C}_2\text{H}_{5,16}^* \) chemiluminescence intensity ratio is the most promising one whilst it showed only slightly changing characteristics with the growing axial distances. The mentioned flame quenching effect by the presence of fuel droplets proposed by Muruganandam et al. [113] has not exactly been observed in the current measurement series, due to the presence of the mixing tube and a subsequent diffuser after it.

The analysis of the chemiluminescence intensity ratios reveals a strongly increasing measurement uncertainty with axial distance, therefore installing a sensor too far from
the burner lip may lead to highly uncertain measurements. Despite the investigation was carried out at 15 kW combustion power, at higher loadings the flame height remains similar, while it is fully turbulent [68]. Therefore, the position of the spectrometer is as satisfactory at idle conditions as it is at full load if the combustion is fully turbulent. It is typical in most of the industrial-scale applications. Based on the measurement results, shown in Fig. 6.6, a chemiluminescence-based equivalence ratio sensor is recommended to be placed in the vicinity of the burner lip to ensure a reliable operation.

6.4 FES of rapeseed oil

![Graphs showing corrected chemiluminescence intensity ratios of OH*, CH*, and C* to C2,516 of rapeseed oil combustion.]

Figure 6.7: Corrected chemiluminescence intensity ratios of OH*, CH*, C2,516 to C2,554 of rapeseed oil combustion.

It was concluded in Section 6.3 that a spectrometer should point just above the burner lip in order to ensure low uncertainty. It was assumed that the device has one intensity unit resolution, which is also the estimated accuracy. Note that the combined expanded uncertainty of the chemiluminescence intensity measurement was omitted in the respective
figures, while the markers overlapped the error bars. Their values of diesel oil and rapeseed oil flames were below 1.4% and 3.0%, respectively at 95% level of significance in all cases.

Figure 6.7 shows the corrected chemiluminescence intensity ratios of OH$^\ast$/C$_{2,554}^\ast$, CH$^\ast$/C$_{2,554}^\ast$, and C$_{2,516}/C_{2,554}^\ast$ as function of λ at different atomizing pressures. On average, all the trends show a strong monotonous increase, which makes the chemiluminescence intensity peak at 554 nm feasible for equivalence ratio estimation in the case of rapeseed oil flames.

Surprisingly, the C$_{2,516}/C_{2,554}^\ast$ is also sensitive to λ, regardless that they are both excited C$_2$ radicals. However, this chemiluminescence intensity ratio is far lower than that of OH$^\ast$/C$_{2,554}^\ast$, CH$^\ast$/C$_{2,554}^\ast$ or C$_{2,516}/C_{2,554}^\ast$ under lean conditions, as it was expected, while these peaks are relatively more intense at less luminous flames.

Regardless that the C$_{2,554}^\ast$ intensity peak in Fig. 6.2 is also identifiable in the case of diesel oil combustion, it completely fades to the background noise as the fuel-air mixture tends to the richer region. Therefore, it is difficult to use this result for equivalence ratio determination generally. A deeper investigation of the background chemistry is suggested in the future to understand this phenomenon.

![Corrected chemiluminescence intensity ratios of OH$^\ast$/CH$^\ast$ of diesel oil (top) and rapeseed oil combustion (bottom).](image)

Figure 6.8 shows the OH$^\ast$/CH$^\ast$ corrected intensity ratio of diesel and rapeseed oil combustion. The former one shows the expected trend, i.e., there is a slight increase in the OH$^\ast$/CH$^\ast$ chemiluminescence intensity ratio above stoichiometric conditions, and a slightly descending trend shows up below that. This behavior was found as well by Docquier et al. [117].

75
Rapeseed oil flames resulted in more flat trends, even at a higher atomizing pressure, showing that the OH*/CH* chemiluminescence intensity ratio is not an appropriate diagnostic property alone in this case. Involving the C*_{2,516}, the ratios of OH*/C*_{2,516} and CH*/C*_{2,516} also show a monotonous increasing behavior on average for rapeseed oil and diesel oil combustion as well, shown in Figs. 6.9 and 6.10. These trends indicate that the OH*/C*_{2,516} and CH*/C*_{2,516} chemiluminescence intensity ratios are also feasible for determining the air-fuel equivalence ratio for diesel fuel combustion at different atomizing pressures. In the case of rapeseed oil combustion, CH*/C*_{2,516} is also suitable. The pressure dependence of OH*/CH* for both fuels can be explained by the different spray structure and mixing. This finding was proposed also by Muruganandam et al. [113]. Higher atomizing pressure results in smaller droplets, which can evaporate in a shorter time, leading to better mixing and generally less luminous flame, as it was concluded earlier. The OH*/C*_{2,516} trends of rapeseed oil combustion are probably related to its low volatility.

![Figure 6.9: Corrected chemiluminescence intensity ratios of OH*/C*_{2,516} of diesel oil (top) and rapeseed oil combustion (bottom).](image)

Except for the OH*/C*_{2,516} chemiluminescence intensity ratio of rapeseed oil combustion and the atomizing pressures of 0.5 bar, all the ratios involving either C*_{2,516} or C*_{2,554} show very low sensitivity to the atomizing pressure compared to the excluded ones. Consequently, in case of fluid fueled flames under low operating pressure, C* plays a significant role which received lower scientific focus compared to either CH* or OH*.

Interestingly, there was no sudden change in the slope of any tendency in chemiluminescence intensity ratio–λ diagrams when the straight diesel oil flame shifted into V-shape.
Figure 6.10: Corrected chemiluminescence intensity ratios of CH$^\ast$/C$_{2,516}^\ast$ of diesel oil (top) and rapeseed oil combustion (bottom).

Note that other variables, such as the pressure, luminosity, and temperature field showed a significant fluctuation. Therefore, this measurement technique was found to be independent from the shape of the flame under all the investigated configurations. Hence, it is a highly potential diagnostic tool in quasi steady-state operation by using long sampling time, i.e., in the order of one second. Here, the integration time was 100 ms and the software averaged 64 profiles to avoid the overloading of the nMOS detector pixels. In this way, a single shoot took 6.4 s. It is known that integration time in the order of one millisecond is short enough to record ongoing unsteady phenomena [109]. Consequently, the measurement uncertainty will also increase.

6.5 Summary of the chapter

In the present chapter, FES measurement of diesel, sunflower, and rapeseed oils were carried out under atmospheric conditions. Firstly, the spectrometer was introduced along with the evaluation methods. A new approach was developed to reduce the measurement noise caused by the black body radiation of the soot particles in the hydrocarbon flame spectra. While crude sunflower oil was found to be stable only at the limitations of the test rig, showing no significant difference in the spectrum compared to diesel oil flame, its further analysis was omitted.

Secondly, the spatial measurement of the flame was carried out at $p_g = 0.35, 0.7,$ and
1 bar and \( \lambda = 1.15 \). It was concluded that the spectrometer should point to the axis, close to the burner lip based on spatial polts of OH*, CH*, and \( \text{C}_2 \text{*}_{2,516} \) and the ratio of them. Regardless that the combustion power may change during operation, the size of fully turbulent flame does not change significantly. Therefore, a fixed sensor is appropriate for such diagnostic techniques.

Thirdly, the comparison of rapeseed and diesel oil spectra was shown, emphasizing a newly identified chemiluminescence intensity peak at 554 nm, which is not yet discussed in the technical literature. This finding may widen the possibilities of flame emission spectroscopy of rapeseed oil flame. The most probable explanation for the occurrence of this peak is the \( \text{A}^3\Pi_g - \text{X}^3\Pi_u \) transition of the excited \( \text{C}_2 \) [152]. By analyzing \( \text{OH}^*/\text{CH}^* \), \( \text{OH}^*/\text{C}_{2,516}^* \), and \( \text{CH}^*/\text{C}_{2,516}^* \) chemiluminescence intensity ratios, it was concluded that \( \text{OH}^*/\text{CH}^* \) is loosely depending on \( \lambda \). However, all ratios including \( \text{C}_{2,516}^* \) have the potential to use for flame diagnostic purposes in the case of turbulent combustion. In addition, the measured intensity ratios were not sensitive neither to the bistable operating regime nor the V-shaped flame versus \( \lambda \). Based on the findings above, the following new scientific results were derived:

The flow patterns and flame shapes of swirling flows are well-documented by Beér [1] and Syred [69]. The varying flame shapes occurred in the measurements are presented in Fig. 5.18. By combining this result with the corresponding spectral analysis in Fig. 6.10, the following new scientific result can be derived:

3. Thesis

In lean premixed prevaporized burners, \( \text{OH}^*/\text{CH}^* \), \( \text{OH}^*/\text{C}_{2,516}^* \), and \( \text{CH}^*/\text{C}_{2,516}^* \) chemiluminescence intensity ratios do not show sudden changes in the function of the equivalence ratio. Even though the flame length, hence its shape, swiftly change due to the breakdown of the precessing vortex core.

The \( \text{A}^3\Pi_g - \text{X}^3\Pi_u \) transition of \( \text{S}(2,3) \) is responsible for the chemiluminescence emission of \( \text{C}_2 \) [152], including the chemiluminescence emission at 554 nm. The chemiluminescence intensity ratios using this peak in the case of rapeseed oil combustion are shown in Fig. 6.7. By considering the newly identified emission peak and the corresponding intensity ratios, the new scientific results can be summarized as:

4. Thesis

During the combustion of crude rapeseed oil under atmospheric conditions and in the investigated air-to-fuel equivalence ratio, \( \lambda = 0.7 - 1.5 \), a significant \( \text{C}_2^* \) chemiluminescence intensity peak was identified at 554 nm, \( \text{C}_{2,554}^* \). Chemiluminescence intensity ratios of either \( \text{OH}^* \), \( \text{CH}^* \), or \( \text{C}_{2,516}^* \) to \( \text{C}_{2,554}^* \) increases monotonically with \( \lambda \).
Conclusions and future directions

Liquid fuel combustion is highly relevant today in both the transportation sector and in the industry. Hence, its four aspects were investigated, namely, atomization, droplet evaporation, combustion, and chemiluminescence emission, using a lean premixed prevaporized burner. Diesel and rapeseed oil utilization was examined in detail, except the case of atomization where only diesel oil spray was measured.

Various formulas exist in the literature to estimate the Sauter Mean Diameter, \( SMD \), of the spray. They are often used even out of their validity range while spray measurement is unfeasible or unaffordable in many circumstances. Therefore, the first aim of the present thesis was to examine several \( SMD \) estimating formulas available in the literature to find out which ones are valid for high-velocity atomization (i.e., from 0.3 up to 3.1 bar atomization gauge pressure). Besides the determination of \( SMD \), the fit of three size distribution functions was analyzed. It turned out that at high atomization pressures the spray is more regular and approaches the \( \Gamma \) distribution function most accurately over the Rosin-Rammler and the Nukiyama-Tanasawa probability density functions.

Regarding evaporation, it turned out that is essential to preheat crude vegetable oils to approach the vaporization characteristics of diesel oil. The increased atomization pressure reduces the evaporation number, \( Ev \), up to the transonic regime. However, the increasing atomization pressure allows less time for evaporation due to the fact that the droplet passes through the mixing tube faster, lowering \( Ev \). Nevertheless, the evaporation time decreases in a higher extent, resulting in an increasing \( Ev \) overall.

It was concluded that rapeseed oil combustion becomes unstable by exceeding the critical swirl number. Neither diffusers, nor confinement, nor elevated combustion power could help the situation. Regarding diesel oil combustion, its blowout stability can be extended by using an appropriate diffuser. In this case, the appearing boundary layer near the diffuser wall provides the improved stability, among the investigated cases, both 15\(^\circ\) and 30\(^\circ\) diffusers performed better by 42% compared to the original case. By using the original burner configuration, combustion of diesel and rapeseed oils were compared from pollutant emission point of view. The atomizing pressure affected both the CO and the NO\(_{X}\) emissions, but both fuels resulted in a similar overall emission when compared to the limitations set by the actual Hungarian decree.

Spatial flame emission spectrometry revealed that the strongest chemiluminescence intensity signal is located just above the burner lip. Beside the well-known chemiluminescence emission of OH\(^*\), CH\(^*\), and C\(_2\)\(_{2,516}\) of hydrocarbon combustion, rapeseed oil flame showed a strong C\(_2\) emission at 554 nm. This peak was found to be suitable for diagnostic purposes. Regardless that the intensity of the mentioned radicals decrease as the flame is either bistable or V-shaped, their ratio hides this information and remain consistent with the earlier trends in the function of air-to-fuel equivalence ratio. Therefore, this technique is suitable for real-time combustion control through equivalence ratio estimation, regardless that the flame is unsteady or even its shape alters.
As for future directions, acoustical monitoring of a flame that may be bistable during the operation of a heat engine is crucial and worth a detailed investigation. Therefore, self-exciting oscillations and other harmful phenomena can be avoided or easier detected and counteracted.

A proper evaporation model would be a huge impact both for CFD and burner designers while still the $D^2$ law is the most reliable tool which incorporates significant restrictions. The inclusion of the thermal radiation and the accurate involvement of turbulent properties would probably result in a more accurate estimation.

Another possible future direction of the present thesis is measuring large-scale burners by flame emission spectrometry. Elevated pressure systems are rarely investigated, despite that state-of-the-art commercial gas turbines operate at the pressure ratio of 50 and beyond.

CFD has proven as an extremely powerful tool for both incompressible and compressible flows. Nevertheless, proper atomization modeling is still a challenging task. Furthermore, simulation of reactive flows is cumbersome with the tools available today for such a purpose. Therefore, validated results by higher hydrocarbon combustion measurements similarly represent high scientific value. The best solution would be a systematic guideline for numerical modeling while it requires significant effort and skill to get an appropriate result. Using detailed chemistry and Direct Numerical Simulation instead of turbulence models or Large Eddy Simulation probably remain impossible in practical scales in the upcoming decades.

A fundamental question what remained unanswered is the source reaction of \text{C}_2\text{,}5\text{,}54. How is it possible to have it to a greater extent in the case of rapeseed oil combustion compared to diesel oil combustion? Why is this scales differently with the equivalence ratio than \text{C}_2\text{,}5\text{,}16? More tests with rapeseed oils of different origin and various crude vegetable oils are required to investigate this phenomenon from the side of measurements. In parallel with this, a detailed reaction kinetic investigation might lead to the theoretical background of this result.
Theses

Based on the findings of Lefebvre [12], Heng et al. [23] and Varga et al. [36], the SMD negligibly depends on the ambient pressure and $SMD \sim We_A^{-1/2}$ at high-velocity atomization. By neglecting the viscosity term in Eq. 2.6, its range of validity is significantly extend as follows:

1. Thesis

In the case of plain-jet airblast atomization of standard diesel oil, the following formula estimates the volume-to-surface diameter, $SMD$, of the spray in the range of Mach number, $Ma = 0.6 - 1.6$ and air-to-liquid mass flow ratio, $ALR = 0.78 - 2.3$:

$$SMD = 0.66d_0 We_A^{-0.5} (1 + 1/ALR),$$

where $d_0$ is the diameter of the liquid jet and $We_A$ is the Weber number based on the properties of the atomizing air. [153–155]

Figures 5.8 and 5.9 compared the blowout mechanism of the original burner with the configuration where a subsequent 45° diffuser was used. By combining this result with the blowout stability characteristics of diffusers with various half cone angle, shown in Fig. 5.11, the following thesis can be stated:

2. Thesis

Blowout stability of V-shaped flames characterized by strong swirl in lean premixed prevaporized burners enhances by applying a subsequent diffuser on the burner lip, which substitutes the outer recirculation zone. In this case, the appearing boundary layer near its wall ensures the improvement of the blowout stability. [156–161]

The flow patterns and flame shapes of swirling flows are well-documented by Beér [1] and Syred [69]. The varying flame shapes occurred in the measurements are presented in Fig. 5.18. By combining this result with the corresponding spectral analysis in Fig. 6.10, the following new scientific result can be derived:

3. Thesis

In lean premixed prevaporized burners, $OH^*/CH^*$, $OH^*/C_{2,516}^*$, and $CH^*/C_{2,516}^*$ chemiluminescence intensity ratios do not show sudden changes in the function of the equivalence ratio. Even though the flame length,
hence its shape, swiftly change due to the breakdown of the precessing vortex core. [161,162]

The $A^3\Pi_g-X^3\Pi_u$ transition of $S(2,3)$ is responsible for the chemiluminescence emission of $C_2^*$ [152], including the chemiluminescence emission at 554 nm. The chemiluminescence intensity ratios using this peak in the case of rapeseed oil combustion are shown in Fig. 6.7. By considering the newly identified emission peak and the corresponding intensity ratios, the new scientific results can be summarized as:

4. Thesis

During the combustion of crude rapeseed oil under atmospheric conditions and in the investigated air-to-fuel equivalence ratio, $\lambda = 0.7 - 1.5$, a significant $C_2^*$ chemiluminescence intensity peak was identified at 554 nm, $C_{2,554}^*$. Chemiluminescence intensity ratios of either $\text{OH}^*$, $\text{CH}^*$, or $C_{2,516}^*$ to $C_{2,554}^*$ increases monotonically with $\lambda$. [161,162]
References


[129] Lemuel M. Diamante and Tianying Lan. Absolute Viscosities of Vegetable Oils at Different Temperatures and Shear Rate Range of 64.5 to 4835 s⁻¹. *Journal of Food Processing*, 2014:1–6, 2014.


Appendix A

Uncertainty and data sheet of measuring devices

The present chapter summarizes the uncertainty of each notable device used during the measurements. The first part details the atomization test rig while the second one specifies the combustion test rig. Note that all the uncertainties are respective to 95% level of significance.

![Diagram of the atomization test rig](image)

**Figure A.1**: The atomization test rig (detailed).

Figure A.1 shows the atomization test rig with numbered parts. Their properties are detailed below.

1. Thermometer
   - Type: Pt100, Class B
   - Measurement range: -200–600 °C
• Accuracy: 0.5 °C

2. Pressure transducer
   • Type: Siemens SITRANS P serie Z
   • Model no.: 7MF1563-5BE00
   • Measurement range: 0–4 bar (absolute)
   • Accuracy: 1000 kPa (0.25% of the maximum)
   • Note: All the other pressure transducers were used for check purposes.

3. Mass flow meter (for fuel)
   • Type: Siemens SITRANS F C Mass 2100 Di 3
   • Measurement range: 0–250 kg/h
   • Accuracy: 0.8% (at 0.35 g/s mass flow rate)

4. Mass flow meter (for air)
   • Type: Omega FMA-A2117
   • Maximum measurement range: 0–100 standard liter/min
   • Used measurement range: 0–50 SLM
   • Accuracy: 1% of the full scale (0.5 SLM)

5. Mass flow meter (for air)
   • Type: Sierra SmartTrak 100
   • Measurement range: 0–1000 SLM
   • Accuracy: 0.5% of the full scale (5 SLM)

6. Traverse system (not shown here)
   • Positioning uncertainty: 0.1 mm

Determining the uncertainty of the PDA system in the case of spray measurement is cumbersome and typically not discussed in the literature. There can be multiple droplets present in the measuring volume which might bias the results due to multiple reflections and light scattering. Therefore, the appropriate phase shift in the light signal, what the PDA detects, is highly noisy. Hence, the inappropriate signals were automatically rejected by the BSA Flow software by Dantec Dynamics. Note that these points were excluded and the collected 20,000 samples at each point means the number of accepted samples. The lowest measured droplet size was 10 nm which is also the resolution of the used 2D Fiber Based PDA by Dantec Dynamics. The uncertainty in the micrometer range is estimated to be 5%. However, it is expected that the system is more accurate (personal communication with Matouš Zaremba, May 18, 2016). Kourmatzis et al. [6] used a spray generator with a nominal droplet size of 40 µm. The SMD values based on the measurements varied between 38 and 45 µm, depending on the operation of the generator. As a consequence, the present thesis neglects the classical uncertainty analysis of the PDA measurement. Due to the good correlation between the adiabatic expansion theory and the measured
thrust of the nozzle, shown in Fig. 4.3, the mass flow rates were estimated based on the pressure measurement. Hence, the uncertainty of the SMD estimation was 4.8%–0.26% at $p_g = 0.3 – 3.1$ bar, respectively. Consequently, all the measured flow rates were used only for check purposes and represented higher uncertainties.

1. Fan
   - Type: Szellőző Művek Budapest, KNV 71
   - Nominal pressure increase: 750–487 Pa (at 540–1290 Nm$^3$/h)
   - Nominal RPM: 2880 1/min
   - Note: Controlled by a continuously adjustable frequency converter.

2. Rotameter 1 (for combustion air)
   - Type: Unirota 93-50-196
   - Measurement range: 3–30 Nm$^3$/h
   - Accuracy: 1.6% accuracy class according to VDI/VDE 3513 guideline

3. Rotameter 2 (for atomizing air)
   - Type: Unirota RM-02 12-468-431
   - Measurement range: 3–30 Nl/min
   - Accuracy: 4.0% accuracy class according to VDI/VDE 3513 guideline

4. Scale (identical ones were used)
   - Type: Radwag WLC 12/30/C1/R
• Accuracy: 0.2 g until 12 kg, 0.5 g until 30 kg.
• Note: During the measurements, 5-liter tanks were used, therefore, the lower accuracy was applied. The scale was purchased for industrial-related projects, but as a consequence, its accuracy was approved by National Office of Measures, Hungary (No.: MKEH 221336 and MKEH 221337).

5. Pressure transducer
   • Type: Siemens SITRANS P serie Z
   • Model no.: 7MF1563-5BE00
   • Measurement range: 0–4 bar (absolute)
   • Accuracy: 1000 kPa (0.25% of the maximum)

6. Combustion air preheater
   • Type: Custom built
   • Accuracy: 1 °C.
   • Control: HAGA KD48P-1200, relay-based
   • Note: It heats the combustion air flow to 400 °C.

7. Fuel preheater (for rapeseed and sunflower oils)
   • Type: Custom built
   • Accuracy: 1 °C.
   • Control: HAGA KD9-h-P, PID
   • Note: It heats the fuel flow to 150 °C.

8. Portable flue gas analyzer
   • Type: Testo 350
   • Order No.: 60253978
   • Accuracy: Listed in Table A.1.
   • Note: Measurement of NO\textsubscript{X}, CO, and O\textsubscript{2}.

9. Spectrometer
   • Type: Custom design by OpLab Ltd.
   • Sensor: Hamatsu nMOS S3904-1024Q
   • Accuracy: Estimated as one intensity unit.

10. Pump 1 (for rapeseed and sunflower oils)
    • Type: Weldon pump C3045-A
    • Control: DC voltage

11. Pump 2 (for diesel oil)
    • Type: Bosch JKB850C3
• Control: DC voltage

12. Pressure regulator

• Type: Rexroth AS3-FRE-6012-GAU FD:208
• Inlet pressure: 6 bar

The VDI/VDE 3513 guideline determines the maximum uncertainty of a rotameter, classifying them by Renard series. Their accuracies can be estimated by the accuracy class, $AC$, as follows:

$$\sigma = AC/4 \cdot \text{range} + AC \cdot 3/4 \cdot \text{reading}. \quad (A.1)$$

Hence, the combined expanded uncertainty of $\lambda$ varies from 3.3%–4.6%. The highest value corresponds to the minimum combustion and atomizing air flow rate.

The spectrometer software, developed by OpLab Ltd., allows stretch and offset of the input signal. Hence, an Avantes AvaLight mercury-argon lamp was used for calibration purposes with well-known spectral peaks. Six different wavelengths were selected in the range of 260-580 nm (296.73, 313.16, 365.02, 404.66, 435.84, 546.07 nm). By setting the above-mentioned two parameters, $R^2$ of 0.999997 was achieved. The highest uncertainty of intensity ratios of rapeseed oil combustion was 3%, but the average value was below 1.5%. As for diesel oil combustion, the maximum uncertainty was less than 1.5%.

Regarding the emission analysis, Table A.1 summarizes the nominal accuracy of each module. The device measured unburnt hydrocarbons, CO$_2$, and SO$_2$ also. However, these results were omitted. In our case, the relatively low CO concentration suggests that the combustion was nearly complete. Therefore, the CO$_2$ measurement does not contain additional information to O$_2$ measurement. The fuels were free from sulfur, as it was mentioned in the introduction. Therefore, no SO$_2$ was detected under the exhaustive measurement series.

<table>
<thead>
<tr>
<th>Measured component</th>
<th>Measurement range</th>
<th>Resolution</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>O$_2$</td>
<td>0-25 V/V%</td>
<td>0.01 V/V%</td>
<td>± 0.2 V/V%</td>
</tr>
<tr>
<td>CO</td>
<td>0-10000 ppm</td>
<td>1 ppm</td>
<td>± 10 ppm (0-199ppm)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>± 5% (200-2000 ppm)</td>
</tr>
<tr>
<td>NO$_X$</td>
<td>0-4000 ppm</td>
<td>1 ppm</td>
<td>± 5 ppm (0-99 ppm)</td>
</tr>
</tbody>
</table>

Table A.1: Accuracy of the monitored modules of the Testo 350 flue gas analyzer.

In order to provide comparable data, all the measured pollutants were systematically modified to 15% O$_2$ level by multiplying the measured value with $(21 - 15)/(21 - O_2)$. While the combustion was carried out under atmospheric conditions, the high amount of entrained air considerably diluted the flue gas. Hence, the measured O$_2$ concentration was between 18.6 and 19.5 V/V%. Therefore, the maximum uncertainty of NO$_X$ and CO measurement were 39 and 79 ppm, respectively. Due to the highly uncertain measurements based on the accuracies given by the manufacturer, another flue gas analyzer, a Horiba MEXA-8120F was used for check purposes. The latter device has an accuracy of 2.1% for NO$_X$ and CO emissions. The two devices showed very close results. In addition, to get quantitative information, the Testo 350 emission analyzer was checked with calibration gasses three times, using one second sampling time for at least two minutes. The results are summarized in Table A.2.
Table A.2: Calibration results of the Testo 350 flue gas analyzer.

<table>
<thead>
<tr>
<th>Pollutant</th>
<th>Nominal concentration [ppm]</th>
<th>Accuracy min/avg/max [ppm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO</td>
<td>187 ppm</td>
<td>0/2.33/3</td>
</tr>
<tr>
<td>NO\textsubscript{X}</td>
<td>95 ppm</td>
<td>0/0.5/2</td>
</tr>
</tbody>
</table>

In the light of these results, the estimated uncertainty of the NO\textsubscript{X} and CO emission analysis are 16 and 47 ppm, respectively. Note that the maximum values were used from Table A.2. The special gasses were provided by Messer and Linde. It is expected that using a closed combustion chamber provides lower uncertainties by the exclusion of the high amount of entrained cold air. However, the goal of pollutant emission analysis was to investigate the effect of different operating conditions; a real burner has to fulfill the corresponding standards and decrees.
Appendix B

Estimation of the swirl number

The present chapter presents an excerpt from the the Combustion Aerodynamics textbook [1] for estimating the swirl number, $S$. In general, the axial flux of the angular momentum, $G_\varphi$, and the axial thrust, $G_z$, can be calculated by Eqs. B.1 and B.2, assuming the conservation of momentum:

\[ G_\varphi = \int_0^R (w_\varphi r) \rho_A w_z 2\pi r dr = \text{const.}, \tag{B.1} \]

\[ G_z = \int_0^R w_z \rho_A w_z 2\pi r dr + \int_0^R p_{\text{sta}} 2\pi r dr = \text{const.}, \tag{B.2} \]

where $R$ is the inner radius of the mixing tube of the burner, shown in Fig. 4.1. $\rho_A$ is the density of the combustion air and $p_{\text{sta}}$ is the static pressure. The swirl number can be expressed as follows:

\[ S = \frac{G_\varphi}{G_z R}. \tag{B.3} \]

In general, the required spatial variables are unaffordable or unfeasible. Therefore, an approximation is used, which role is to simplify Eq. B.1 and eliminate the pressure term in Eq. B.2 by calculating $G_z$ at the inlet:

\[ G'_z = 2\pi \int_0^R \rho w_z^2 r dr. \tag{B.4} \]

Figure B.1: The cross section of the swirl vanes.
$G_{\phi}$ is rather geometry dependent. The presently discussed case is shown in Fig. B.1, showing the main dimensions. Here, $G_{\phi}$ is estimated by Eq. B.5 at the inlet, too. The detailed steps of its derivation is omitted now.

$$G_{\phi} = \frac{1}{1 - \psi} \frac{\tan \alpha}{1 + \tan \alpha \tan(\pi/Z)} \frac{\dot{m}^2}{\rho 2\pi N},$$  \hfill (B.5)

where $N$ is the axial length of the swirl vanes and $\psi$ is the blockage factor which be calculated by Eq. B.6 for vane cascade:

$$\psi = \frac{Zs}{2\pi R \cos \alpha},$$  \hfill (B.6)

where $Z$ is the number of swirl vanes and $s$ is the width of a single vane at the inner side of the mixing tube. Hence, the modified – or geometric – swirl number, $S'$, can be calculated now:

$$S' = \frac{G_{\phi}}{G'zR}.$$  \hfill (B.7)

This is the swirl number what is used in the literature extensively. The present thesis is confined to the modified swirl number instead of the generalized one. For simplification, the modified swirl number is noted with $S$ instead of $S'$ in the text to be consistent with the technical literature.
Appendix C

Detailed fuel properties for calculating evaporation

Since Eqs. 2.21–2.28 contain temperature-dependent variables, this section details the used functions and constants for the investigated fuels, used in Chapter 4. Firstly, the calculation of latent heat is detailed which is followed by the material property functions of diesel oil and n-heptane based on the Atomization and Sprays textbook by Lefebvre [13]. Next, the properties of ethanol are shown, gathered from high-quality web databases [137–140].

Lastly, the properties of rapeseed and soybean oils are detailed [135,141,143–145]. The respective sources are also cited above the equations and tables. Note that the thermal conductivity and specific heat capacity at constant pressure of air were based on the NIST database [137] and the work of Lemmon et al. [163]. Their values at the desired temperature were calculated by linear interpolation.

The temperature-dependent latent heat was calculated by Eq. C.1 [13]:

\[
L = L_{T_b} \left( \frac{T_c - T_{s,hu}}{T_c - T_b} \right)^{-0.38},
\]

where \(T_b\) is the boiling temperature and \(c\) subscript denotes the critical value. The required material properties are listed in Table C.1.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>(T_c) [K]</th>
<th>(T_b) [K]</th>
<th>(L_{T_b}) [kJ/kg]</th>
<th>(M_L) [kg/kmol]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>725.9</td>
<td>536.4</td>
<td>254</td>
<td>198</td>
</tr>
<tr>
<td>Ethanol</td>
<td>517.15</td>
<td>351.52</td>
<td>906</td>
<td>46.07</td>
</tr>
<tr>
<td>n-Heptane</td>
<td>540.17</td>
<td>371.4</td>
<td>371.8</td>
<td>100.2</td>
</tr>
<tr>
<td>Rapeseed</td>
<td>765.15</td>
<td>584.15</td>
<td>209</td>
<td>926</td>
</tr>
<tr>
<td>Soybean</td>
<td>775.0</td>
<td>606.4</td>
<td>211.3</td>
<td>875</td>
</tr>
</tbody>
</table>

Table C.1: Fuel properties at 21 °C [13,138,141,146].

The vapor pressure calculation of n-heptane and diesel oil is based on the Clausius-Capeyron equation. The following form was used for diesel oil and n-heptane [13]:

\[
p_{L,s} = e^{C_1 - C_2/(T_s - 43)}/100,
\]

where \(C_1\) and \(C_2\) are constants. The result is divided by 100 while it gives the vapor pressure in kPa dimension and bar is used presently. The constant values are summarized in Table C.2, respective to \(T_b\).
The specific heat capacity at constant pressure of diesel oil and n-heptane were calculated by using Eq. C.3 for the liquid and the gaseous phase [13].

\[
c_{p,L} = \begin{cases} 
760 + 3.35T/\sqrt{0.001\rho_L}, & \text{if the phase is liquid,} \\
(363 + 0.467T)(5 - 0.001\rho_{288}), & \text{if the phase is gasous,} 
\end{cases} 
\]  

(C.3)

where the 288 subscript denotes the property at 288 K. The density of diesel oil and n-heptane in liquid phase was calculated by Eq. C.4.

\[
\rho_L = \rho_{288} \left[ 1 - 1.8C_3(T - 288) - 0.09\frac{(T - 288)^2}{(T_c - 288)^2} \right],
\]  

(C.4)

where \(C_3\) is a constant, calculated as follows:

\[
C_3 = 5.432 \cdot 10^{-12}\rho_{288}^3 - 1.113 \cdot 10^{-8}\rho_{288}^2 + 6.011 \cdot 10^{-6}\rho_{288} + 6.316 \cdot 10^{-5}. 
\]  

(C.5)

The thermal conductivity of diesel oil and n-heptane vapors were calculated by polynomials based on Eqs. C.6 and C.7, respectively [13].

\[
k_L = -2.441 \cdot 10^{-10}T^3 + 4.888 \cdot 10^{-7}T^2 - 1.884 \cdot 10^{-4}T + 0.029 
\]  

\[
k_L = -2.604 \cdot 10^{-10}T^3 + 5.187 \cdot 10^{-7}T^2 - 2.008 \cdot 10^{-4}T + 0.033 
\]  

(C.6)  

(C.7)

The used Clausius-Capeyron equation is slightly different in the case of ethanol, based on the National Institute of Standards and Technology (NIST) database [137]:

\[
\rho_{L,s} = \frac{C_4}{C_6 - C_4};
\]  

(C.8)

where \(C_4\), \(C_5\), and \(C_6\) are constants. Their values are given in Table C.3.

<table>
<thead>
<tr>
<th>Ethanol</th>
<th>(C_4)</th>
<th>(C_5)</th>
<th>(C_6)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T &gt; T_b)</td>
<td>4.92531</td>
<td>1432.53</td>
<td>61.819</td>
</tr>
<tr>
<td>(T &lt; T_b)</td>
<td>5.37229</td>
<td>1670.41</td>
<td>40.191</td>
</tr>
</tbody>
</table>

Table C.3: Constants of ethanol vapor pressure [137].

The specific heat capacity at constant pressure in liquid and vapor form is calculated by Eq. C.9 [137,139].

\[
c_{p,L} = \begin{cases} 
4.405 \cdot 10^{-5}T^2 - 0.018T + 3.934, & \text{if the phase is liquid,} \\
3 \cdot 10^{-7}T^3 - 0.0018T^2 + 4.2895T + 312.29, & \text{if the phase is gasous,} 
\end{cases} 
\]  

(C.9)
Equations C.10 and C.11 were used for calculating the heat conductivity of ethanol vapor and the liquid density, respectively [138,140].

\[ k_L = 8 \cdot 10^{-8}T^2 + 5 \cdot 10^{-5}T - 0.009 \]  \hspace{1cm} (C.10)

\[ \rho_L = 99.3974/0.310729^{1+(1-T/513.8)^{0.305143}}. \]  \hspace{1cm} (C.11)

The vapor pressure at the drop surface of rapeseed oil was calculated by Eq. C.12 which is an Arrhenius-type relation instead of the previously used Clausius-Capeyron equations [141]:

\[ p_{L,s} = C_7e^{-C_8/T}, \]  \hspace{1cm} (C.12)

where \( C_7 \) and \( C_8 \) constants were calculated by Eqs. C.13 and C.14:

\[ C_7 = 1.013e^{\frac{L_T \cdot M_L}{R_u \cdot T_b}}, \]  \hspace{1cm} (C.13)

\[ C_8 = L_T \cdot M_L / R_u, \]  \hspace{1cm} (C.14)

where \( R_u = 8.314 \) kJ/K kmol is the universal gas constant. The specific heat, thermal conductivity, and the density were calculated using Eqs.C.15 –C.17 [135,141,143]:

\[ c_{p,L} = 1000 \left[ 0.003003 \left( T - 273 \right) + 2.086 \right], \]  \hspace{1cm} (C.15)

\[ k_L = 0.15 \sqrt{T/373}, \]  \hspace{1cm} (C.16)

\[ \rho_L = 1000 \left[ 0.9273 - 0.0061837 \left( T - 273 \right) \right]. \]  \hspace{1cm} (C.17)

As for soybean oil, the vapor pressure at the surface, specific heat capacity at constant pressure, heat conductivity, and density were calculated by Eqs. C.18–C.21 [135,144,145]:

\[ p_{L,s} = 2.24 \cdot 10^{-11}e^{0.040006T}, \]  \hspace{1cm} (C.18)

\[ c_{p,L} = 1000 \left[ 0.002792 \left( T - 273 \right) + 1.956 \right], \]  \hspace{1cm} (C.19)

\[ k_L = 0.194 \cdot 10^{-4} \left( 1119.3 - 0.68T \right)^{4/3}, \]  \hspace{1cm} (C.20)

\[ \rho_L = 1115 - 0.633T. \]  \hspace{1cm} (C.21)
Appendix D

Blowout characteristics

The flame blowout mechanism was shown earlier in Figs. 5.8 and 5.9. The role of this part is to present the sequences from a high-speed video. While the same camera was used for recording both normal and high-speed videos, the below-presented figures show a different blowout under the same conditions. Due to the 480 frames per second recording mode, the images were inverted to highlight the flame. These sequences are the last frames before the blowout. The resolution of the captured high-speed videos were 224 x 168 pixels.

Figure D.1 clearly demonstrates that the flame detaches from the burner lip, lifts up, and the reaction stops while the recirculation zone was not able to transfer enough heat back to the upstream mixture. But frame 10 shows that some portion of the combustible mixture was relighted, its energy was insufficient to sustain the combustion process.

Figure D.1: Blowout without stabilizer. Sequences from high speed video with inverted colors. The teal lines indicate the burner lip.

Figure D.2 clearly shows the fact that the flame was not lifted up, it was sustained near the diffuser wall until blowout. The preceding low-frequency fluctuation [65] is clearly observable. Considering the filming speed, it is about 120 Hz while the third, the seventh, and the eleventh frame show less luminosity than the preceding and forthcoming frames. However, a microphone can accurately identify the corresponding frequency. If this is
combined with a phase-locked chemiluminescence measurement, the Rayleigh criterion can be checked. Namely, if the heat release and the pressure fluctuations are in phase, their amplitude grows and facilitates flame blowout.