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Combustion Noise from Non-premixed and Lean-premixed Swirl Flames

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Summary
The development of modern gas turbines and jet engines is focused on the reduction of pollutant emissions and, increasingly, on the reduction of overall noise emission, including combustion noise. This requires the minimization of the noise sources, namely noise from the turbulent flow, combustion noise and noise caused by periodic instabilities and fluctuations of the ignition zone. This has to be achieved under conservation of the benefits of swirl flames, e.g. high ignition stability and broad operation ranges. This paper is focused on the description and characterisation of fluctuations of the ignition zone due to different mixing and stabilization characteristics of noise from swirl flames. Another goal was the description of flame noise caused by the formation and burning of coherent, periodic flow structures, often detected in swirl flows and flames [1, 2, 3, 4]. The physical model offers the possibility to understand the formation mechanism of the coherent flow structures and their effect on the combustion of non-premixed and premixed swirl flames.

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1. Introduction

Most technical combustion systems use turbulent premixed or non-premixed swirl flames with high volumetric reaction densities. To realize good ignition stability, especially, when using high air equivalence ratio λ to prevent thermal caused NO_x-emissions, the flow field is swirl stabilized. The swirl flow forms - when a critical value of the swirl number is exceeded - a central inner recirculation zone and causes a longer residence time of reactive species and enables the formation of short flames of high reaction densities with benefits of high ignition stability due to the recirculation of hot gases [5, 6] and achievable low-pollution combustion. Another advantage of premixed swirl-stabilized flames is the option to reduce pollutant emissions of modern combustion systems materialized in industrial, traffic, power plant and aircraft applications compared to diffusive flames by cost- efficient design solutions. Basic research investigation of swirl flames is given by [7, 8].

Besides the named properties of swirl flames, also some issues arised putting this type of burner into action. Swirl flows and flames tend to generate periodic flow instabilities, which lead to an increasing noise emission and problem of flame stability. Combustion instabilities have been investigated intensively in the recent years using pulsated inflows [9, 10]. Instabilities in swirl flames have been investigated by [11, 12, 13] and reveal the importance of coherent structures in the forward flow surrounding the inner recirculation zone. The existence and the influence of coherent structures on the noise emission of the flame and, thus, of the combustion system requires more knowledge to minimize combustion noise in modern industrial and aircraft applications already during the design process. Another disadvantage of swirl flames are their high turbulence-caused noise levels caused by high reaction densities and amplified by the use of premixed flames in industrial furnaces.

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2. Experimental setup

The investigations were carried out using a modular double-concentric swirl-burner for premixed and non-premixed combustion of gaseous fuels, offering the options to vary independently geometrical parameters (especially the angle of the burning gas injection and the burner exit geometry with a recessed pilot often used in industrial applications [14, 15]) as well as technically relevant operation parameters like thermal load $\dot{Q}_{\mathrm{th}}$ and air equivalence ratio $\lambda = \dot{V}_{n,\mathrm{air}}/\dot{V}_{n,\mathrm{fuel}} l_{\min} = 1/\Phi$ with $l_{\min}$ as minimal, stoichiometric required air rate (n: normalized). Another advantage of the burner design is the possibility to generate periodic flow instabilities by variation of the burner exit geometry and definite conservation of the mean turbulent flow field [3]. Identical frequencies have been determined in the spectral distributions of the noise far sound field measured by a microphone probe and in the flow field detected via laser diagnostics (Laser Doppler Velocimetry).

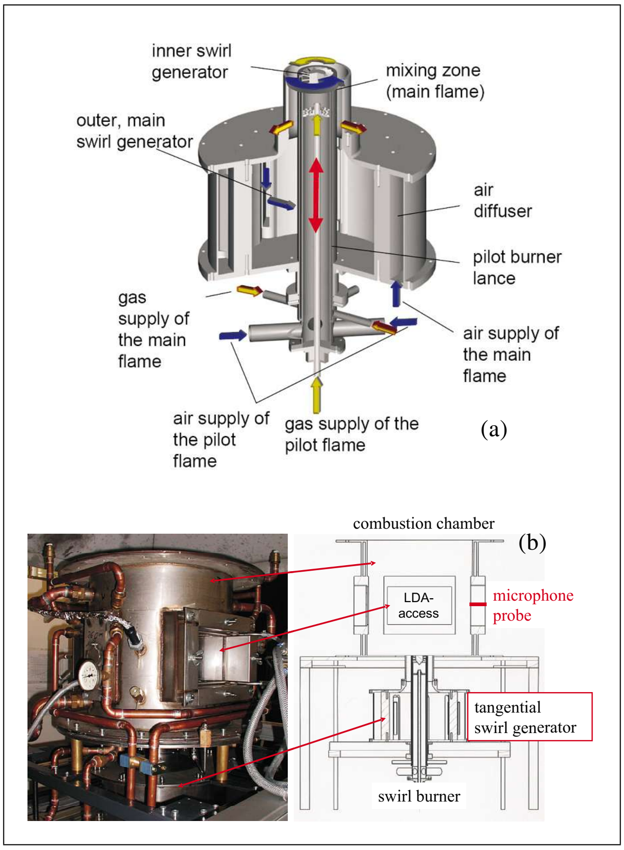

Figure 1. a: Sketch of the double-concentric swirl burner with a central pilot burner. b: Combustion system for investigations under enclosed conditions.

The part of noise caused by the underlying physical mechanism [4] can be quantified and avoided by proper design of the burner outlet geometry.

The influence of the gas injection angle in the case of non-premixed flames was investigated with variation of gas injection angles between 0° and 90° referring to the burner's axis (resulting in type-I and type-II-diffusion flames, see Figure 7, [16]). The operating parameters' range of the pilot burner varied between premixed conditions - leading to sufficient main flame stabilization at the pilot burners exit - and diffusion-type main flame without pilot flame, causing a fluctuation of the ignition zone in dependence of the gas injection angle of the main gas supply and a strong corresponding increase of the flame noise.

Figure 1a shows a sketch of the double-concentric burner, which produces two concentric swirl-flows: A central pilot burner flow and an outer main burner flow. The inner diameter of the main burner nozzle is $D_{0,\text{nozzle}}$ = 110 mm and the outer diameter of the pilot burner is $d_{0,\text{pilotburner}} = 70$ mm. For the isothermal investigations, the inner and outer flows were fed with air from a compressor, the measurements with swirl flames were carried out with premixed or non-premixed natural gas/air mixtures.

$$S_{0,\text{th}} = \frac{\dot{D}}{\dot{I}_{\text{ax},0} R_0}. \tag{1}$$

The normalized momentum flux ratio representing theoretical swirl number $S_{0,\text{th}}$ at the burner's exit is given in equation (1) with the tangential momentum flux $\dot{D}$, the axial momentum flux $\dot{I}_{\text{ax},0}$ and the burner's exit radius R_0.

The swirl of the pilot flame was generated by an axial swirl generator with vanes and a theoretical swirl number $S_{0,\text{th,pilot}} = 0.79$ (equation 1). The main flow was swirled by tangential swirl generators with a theoretical swirl number $S_{0,\text{th,main}} = 0.90$. The axial pilot burner position with respect to the main burner outlet is the variable geometrical parameter. In Figure 1a, the pilot burner outlet is aligned to the burner outlet ($x_{\text{pilotburner}} = 0$ mm). The position of the inner pilot burner can be set up to 40 mm behind the burner outlet ($x_{\text{pilotburner}} = -40$ mm).

The microphone probe was positioned at a fixed measurement position with a radial distance to the center line $r/D_{0,\text{nozzle}} = 4.55$ and an axial distance to the burner outlet of $x/D_{0,\text{nozzle}} = 1.00$, after having proven the independence of the noise measurements from the probe position due to the spherical noise emission characteristics in the near field of the flame [17].

For the investigations of the enclosed flames the double-concentric swirl burner (Figure 1a) was adapted to a water-cooled combustion chamber with an inner diameter D_{cc} = 0.44 m and a length $l_{cc} = 0.5$ m (see Figure 1b).

3. Acoustic characterisation of isothermal swirl flows

In the following section the results of experimental investigations of the isothermal, turbulent flow field of the new burner design - including the options of independent variation of operating parameters and geometrical details - are presented. The experiments were carried out under variation of the main operation and geometrical parameters like swirl intensity, flow rate and burner outlet geometry (axial position of the pilot burner).

In Figure 2 the results of the time-averaged isothermal flow field at four traverses in the vicinity of the burner are plotted. The theoretical swirl number of the main and pilot flow are $S_{0,\text{th,main}} = 0.90$ and $S_{0,\text{th,pilot}} = 0.79$ and the axial position of the pilot burner is $x_{\text{pilotburner}} = 0$ mm. The profiles of the axial velocity, starting nearby the burner outlet ($x/D_0 = 0.05$, cross) with the divided pilot- and main flows, are shown. With increasing distance to the burner outlet the typical behavior of a strongly swirling flow was observed: The separation of the pilot and main flow disappears, indicating the fast mixture of the two flow fields, and - at the centre line - a inner recirculation zone with negative axial velocity (opposite to the main flow direction) can be identified.

The remaining single swirling jet spreads to radial direction and, caused by the free flow field, entrains surrounding, non- rotating medium, while propagating further downstream ($x/D_{0,\text{nozzle}}$ = 0.50, 1.00, 1.50). Measurements of the noise emissions of the isothermal swirl flow showed values of the sound pressure level - calculated with equation (2) - SPL = 75.7 dB ($x_{\text{pilotburner}} = 0$ mm,

$p_{0,\mathrm{rms}} = 2 \cdot 10^{-5}$ Pa) at the fixed measurement position with a radial distance to the center line $r/D_0 = 4.55$ and an axial distance to the burner outlet of $x/D_0 = 1.00$.

$$\mathrm{SPL} = 20 \log \left(\frac{p_{\mathrm{rms}}}{p_{0,\mathrm{rms}}} \right). \tag{2}$$

An interesting result is the negligible influence of the swirl intensity ($0 < S_{0,\mathrm{th}} < 0.9$) and the strong influence of the flow rate (burner outlet velocity) on the turbulent flow noise. The value of 75 dB at a volume flow rate of $\dot{V}_{n,\mathrm{air}}$ = 200 m_n^3/h at the described measurement position is the minimal order of magnitude, which cannot be undercut for the used isothermal, turbulent swirl flow.

3.1. Periodical flow instabilities in isothermal swirl flows

To investigate the dynamic behaviour of the emitted noise more detailed, the acquired data from the microphone probe ($r/D_0 = 4.55$, $x/D_0 = 1.00$) were analysed in the frequency domain to get more information of the spectral distribution of the sound pressure amplitudes. Referring to the first burner configuration (see section 3) the pilot burners outlet was aligned to the burner outlet ($x_{\mathrm{pilotburner}}$ = 0 mm) and the spectral distribution shows the typical behaviour of the turbulent disturbances in an isothermal, turbulent flow field (Figure 3 grey line; [2, 3]).

In Figure 3 the distribution of the emitted sound pressure is shown with the pilot burner drawn back 40 mm upstream the burner outlet (black line). Industrial burner applications with lean-premixed flames often use recessed components like pilot burners and bluff-bodies to prevent thermal damage of the sensitive units, examples are Siemens V64 [14] and Pratt & Whitney FT 8 [15]. The spectral distribution of the sound pressure shows a dominating peak at a frequency $f = 56$ Hz.

In comparison to the aligned pilot burner $x_{\mathrm{pilotburner}}$ = 0 mm the peak increases the total SPL from 75.7 dB to a level of 80.5 dB. The influence on the acoustic field was identified as a consequence of periodic oscillations of the axial position of the inner recirculation zone above the bluff body in the swirl flow and the quantified increase of the sound pressure level is caused by a periodic disturbance of the isothermal flow field, shown by measurements of the flow field and spectral distribution of the axial velocity component. The underlying physical mechanism is described by [13, 3] and shows a time-dependent, periodical fluctuation of the axial position of the recirculation zone, causing a time-dependent effective swirl number $S_{\mathrm{eff}} = S_{\mathrm{eff}}(t)$ and the position of the recirculation zone changes the different positions nearly sinusoidally.

The variable burner design enables to generate periodic flow disturbances in an isothermal swirl flow. The consequence for further burner constructions is the need to prevent coherent flow structures caused e.g. by recessed pilot burner constructions, as often used in common industrial burner designs [14, 15].

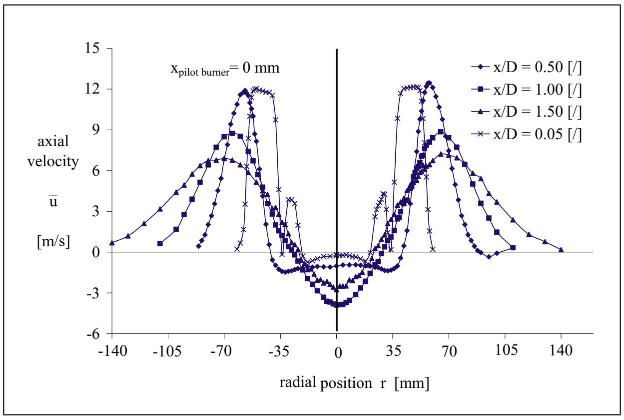

Figure 2. Axial velocity profiles of the isothermal flow at four traverses.

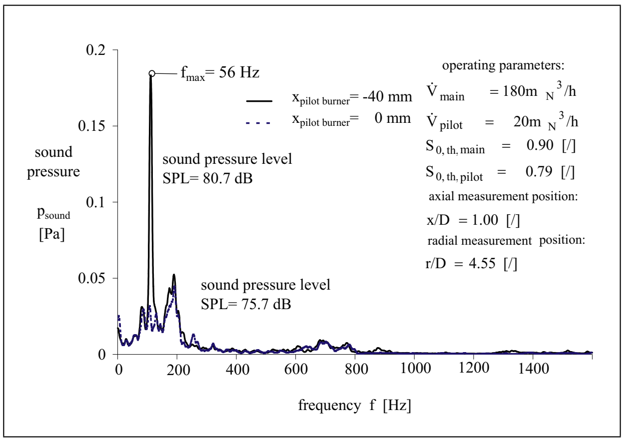

Figure 3. Spectral distribution of the sound pressure of an isothermal swirl flow with different positions of the central pilot burner ($x_{\mathrm{pilotburner}}$ = 0 mm/−40 mm).

4. Premixed swirl flames

4.1. Noise sources in premixed swirl flames: Influence of the operating parameters

In Figure 4 the results of sound pressure measurements of premixed swirl flames are presented. The experiments were carried out with a constant pilot flame of a thermal load $\dot{Q}_{\mathrm{th,pilot}}$ = 20 kW and an air equivalence ratio of $\lambda_{\mathrm{pilot}} = 1.05$ to guarantee a stable pilot flame. The pilot burner is positioned at the level of the burner outlet ($x_{\mathrm{pilotburner}}$ = 0 mm). It is obvious, that the thermal output of the premixed main flame increases the sound pressure considerably. The reacting flow shows a sound pressure level of about 25 dB higher than the SPL of the isothermal flow (gas flow rate substituted by air). This part is called flame-induced noise and is caused by the volume expansion and density fluctuations in the flame due to the massive increase of the exhaust gas temperature. A two times enhanced thermal output of the main flame ($\overline{\dot{Q}}_{\mathrm{th,main}}$ = 150 kW and 300 kW) causes an increase of the SPL from 103 dB to nearly 108.5 dB, whereas the influence of the air equivalence ratio λ_{main} is negligible (see Figure 4).

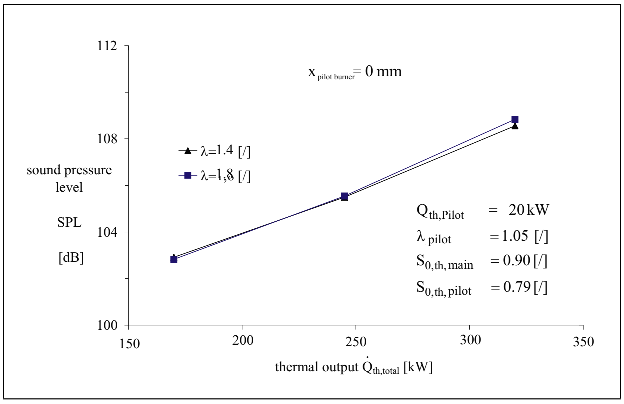

Figure 4. Sound pressure level of premixed flames under variation of thermal load $\bar{\dot{Q}}_{\mathrm{th,main}}$ and air equivalence ratio λ_{main} of the main flame.

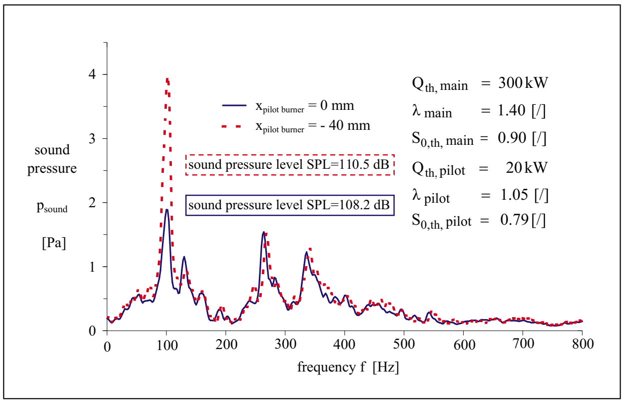

Figure 5. Spectral distribution of the sound pressure of premixed flames with different burner exit geometries (different positions of the pilot burner $x_{\mathrm{pilotburner}} = 0\,\mathrm{mm}$ and -40 mm).

4.2. Noise sources in premixed swirl flames: Variation of geometrical parameters

Next the influence on the noise of premixed swirl flames of the geometrical parameters was investigated. For isothermal flows the influence was predominant (see section 3.1) and the question to be answered was, how premixed flames react on such disturbances of the flow field. In Figure 5 the dependency of the burner outlet geometry on the sound pressure of premixed flames is presented. The spectral distribution is almost identical for all frequencies except $f = 101$ Hz. The recessed pilot burner generates dynamic, periodic structures in the flow field. It should be pointed out that their existence is not limited to isothermal flows. Corresponding to that fact such structures influence the investigated flames and their formation and combustion lead to an increase of the SPL of 2.3 dB (+30%).

Comparing reacting and isothermal cases shows that in the isothermal flow the frequency of the periodic disturbance is lower. The frequency distribution of flames covers a wider frequency band. The Strouhal number Str (equation 2) for both cases reveals the same values Str = 0.74 [/] (premixed flame) and Str = 0.72 [/] (isothermal flow). This proves that the effect is originated from the same physical phenomenon [3].

$$\mathrm{Str} = \frac{f_{\mathrm{peak}} D_0}{\bar{u}_0} \quad [/]. \tag{3}$$

In Figure 6 the results for identical combustion parameters with variation of the burners exit geometry are plotted. The varied pilot burner position influences the sound pressure level of the flame in the same way as in the isothermal case. The recessed pilot burner $x_{\mathrm{pilotburner}} = -40\,\mathrm{mm}$ generates coherent structures in the flow field and, therefore, the SPL increases in the presented case about 2.5 dB (+33.5%).

5. Noise sources in non-premixed swirl flames

In the second part of the characterization and quantification of noise from reacting flows of non-premixed flames,

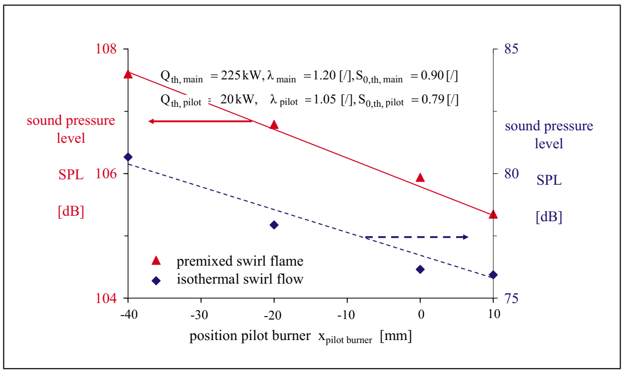

Figure 6. Influence of the pilot burner position $x_{\mathrm{pilotburner}}$ on the sound pressure level of premixed swirl flames.

with the option of using a premixed pilot-flame, were studied and results were compared to investigations of premixed flames, which were carried out to get a classification of all technically relevant type of flames. The difference to the above described premixed case is the location and the type of gas injection. In the case of premixed flames a mixture of fuel and air leaves the burner exit, whereas the air equivalence ratio is constant across the burner outlet area. For non-premixed flames the burner outflow consists of pure air, the gas is injected directly into the flame zone. This leads to a mixture of fuel, air and hot combustion products in the flame zone with the effect of fuel-rich and fuel-lean regions and the effect that the flames are often longer than premixed flames of same operating parameters, caused by a time delay for mixture formation before ignition occurs. Consequently, a lower volumetric reaction density is characteristic for non-premixed flames, which causes disadvantages like higher, local temperatures (NO_x-formation) and soot formation.

5.1. Variation of geometry-parameters

The investigations of non-premixed flames were carried out with a double-concentric swirl burner configuration, mentioned before. The gas injection was located at the central bluff- body of the pilot burner and the parameter to

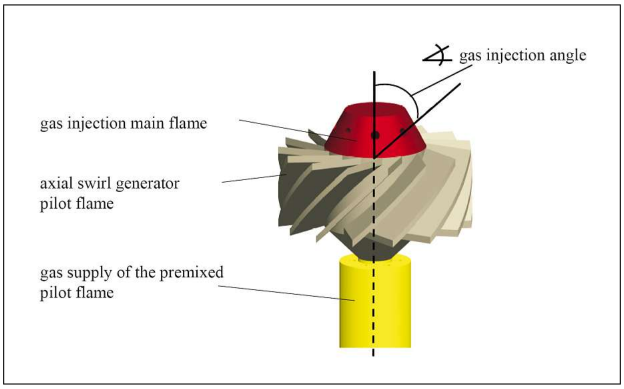

Figure 7. Sketch of the gas injection for non-premixed flames.

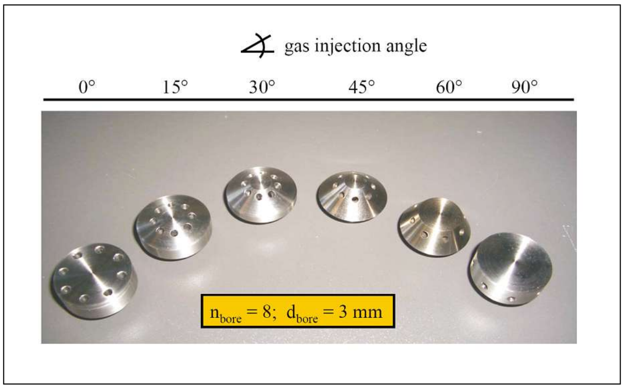

Figure 8. Gas injection nozzles to realize different types of non-premixed flames.

influence the flame characteristics (type-I- / type-II-flame [16]) is the gas injection angle to generate technically relevant flames for different applications like jet engines or industrial furnaces. The angle (see Figure 7) is defined between the direction of the gas flow and the burner axis. Figure 8 shows the realization of different types of non-premixed flames by varying the gas injection angles in a range of 0° to 90° in steps of 15°.

Figure 9 shows the results for variation of gas injection angles between 30° and 90° with variation of the air equivalence ratio. A decrease of air equivalence ratio should induce an increase of sound pressure level (see section 4). But non-premixed flames show a remarkable behaviour: The sound pressure level is almost constant at SPL between 104 dB and 108 dB. A finding is that the non-premixed flames are partly louder than the equivalent premixed flame. The question is, if there are other physical mechanisms influencing the combustion noise. A difference between the operating conditions of premixed and non-premixed flames is the pilot flame. The non-premixed flames have one central fuel supply and the different gas injection angles effect different stabilization points of the different flames. The next consequent step was the prevention of these fluctuations of the stabilization point by use of a premixed pilot flame.

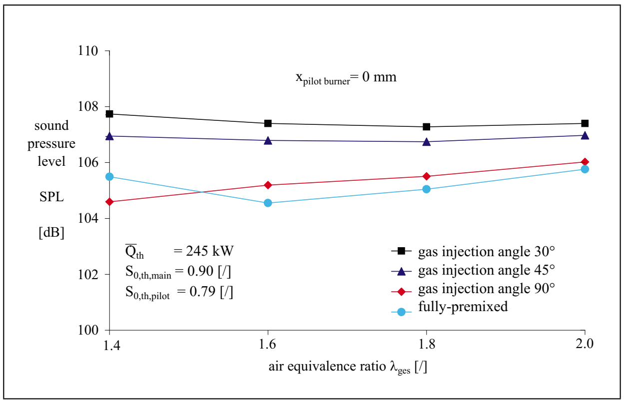

Figure 9. Sound pressure level of non-premixed flames under variation of air equivalence ratio λ_{main} and the gas injection angle.

5.2. Influence of the pilot flame on noise generation

To investigate the influence of main flame stabilization by a premixed pilot flame, the same thermal load and the air equivalence ratio of the pilot flame as adjusted for investigations under premixed conditions of pilot and main flame were used. This enables to compare the premixed and non-premixed main flames with a stable, stationary ignition point of the main flame. The investigations showed the need of a locally stable ignition for investigations of the combustion noise, because the fluctuation of the flame ignition point causes an increase of noise up to 5 dB. But for such conditions a comparison with premixed flame cannot be done. Applying the premixed pilot flame in all cases, the non-premixed flames are always quieter than the comparable premixed flame, which can be explained by lower reaction density of the non-premixed flames.

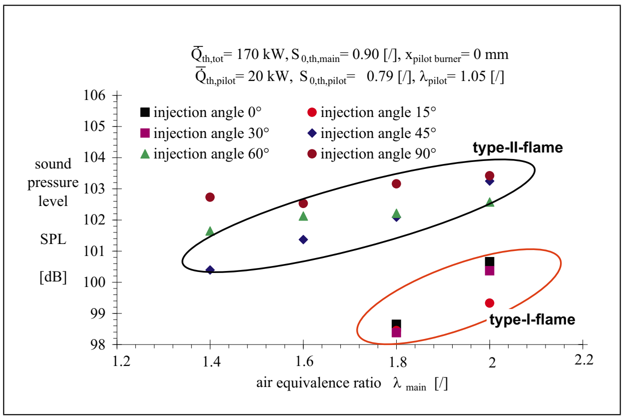

Figure 10. Sound pressure level of non-premixed flames with premixed pilot flame under variation of air equivalence ratio λ_{main} and the gas injection angle.

The experiment showed two regimes of non-premixed flames (Figure 10): Louder flames have an injection angle between 45° and 90° and look like premixed flames (blue, short flame zone) and the SPL are with values of 101–103 dB very close to premixed flames (SPL = 103–105 dB). The loudest non-premixed flames are the flames of a gas injection angle of 90°, leading to almost the combustion noise of fully-premixed flames (Figure 11). The second regime with the smaller injection angles (0–45°)

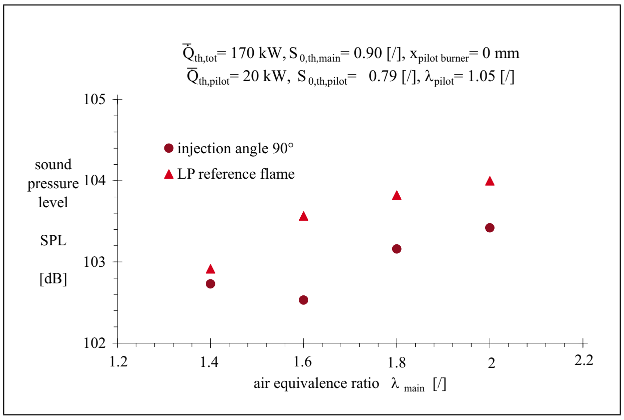

Figure 11. Comparison of the sound pressure level from premixed and non-premixed swirl flames.

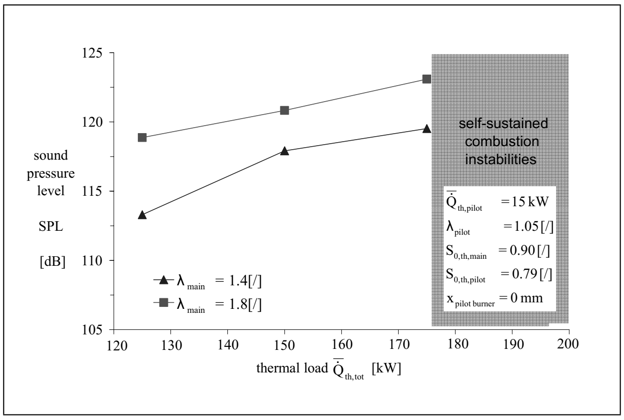

Figure 12. Combustion noise of enclosed, premixed swirl flames.

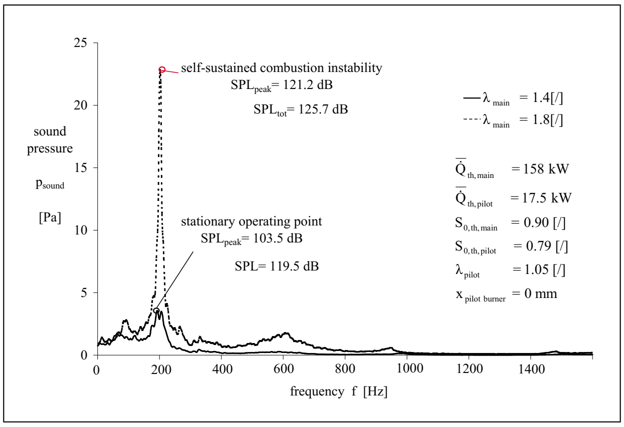

Figure 13. Boundary between combustion noise and self-sustained combustion instabilities.

is characterized by long, soot-forming flames with very low volumetric reaction densities. The SPL of this regime is very low at values of 96–100 dB (dependent from air equivalence ratio λ, see equation 1).

With the stable and time-independent ignition point of the non-premixed flame a possibility is shown to compare different types of non-premixed flames and to compare results with experiments for premixed operating conditions and another physical mechanism of noise generation especially under non-premixed conditions was detected and can be separated from the other noise sources.

6. Enclosed premixed swirl flames

The last step to get a complete understanding of combustion noise sources was the investigation of enclosed swirl flames. The test facility consists of a double concentric swirl burner (Figure 1a) and a combustion chamber with an inner diameter D_{cc} = 0.44 m and a length l_{cc} = 0.5 m (Figure 1b). Figure 12 shows the dependence of the sound pressure level of combustion noise - measured at the wall of the combustion chamber (axial position x_{micro} = 0.25 m, radial position r_{micro} = 0.25 m-caused by the necessity of a water-cooled adapter to prevent thermal damage of the microphone probe) - from the thermal load of the flame and the air equivalence ratio of the main flame (see section 2). The results showed that the operating parameters of the flames are limited, caused by the tendency of lean-premixed flames to get self-sustained combustion instabilities, being an order of magnitude louder than the stationary flames. In Figure 13 the characteristics of a stationary and an oscillating flame are compared and the difference which is limited to the increase of one preferential frequency - leading to an increase of the SPL of more than 6 dB - is obvious. The investigations were made for steady-state operating point to be able to compare it to the results of the non-enclosed premixed flames (see section 4). The most important result is the level of the combustion noise into the chamber which is up to 20 dB louder caused by reflection of noise at the combustion chamber's wall and the higher level of temperature into the chamber with the formation of outer recirculation zones.

7. Conclusions

The investigations shown in the presented work cover detailed noise measurements of isothermal swirl flows and swirl flames (fully-premixed and non-premixed). The spectral analysis of pressure fluctuations shows explicit preferential frequencies for the isothermal and the reacting case (premixed conditions). As the source, coherent structures were detected. The underlying physical mechanism [13, 4] is the same for the isothermal and reacting flow.

The results of detailed noise measurements enable to correlate the total sound pressure level of swirl flames with its sources namely, turbulent flow (75 dB), combustion noise (+25 dB) and coherent flow structures (+2.5 dB), the latter being caused by an unfavorable burner exit geometry, that is, unfortunately, very common in industrial burner designs. The formation and reaction of these flow structures lead to an increase of SPL up to 5 dB for isothermal flow and up to 3 dB for swirl flames. Their prevention results in a significant decrease of combustion noise.

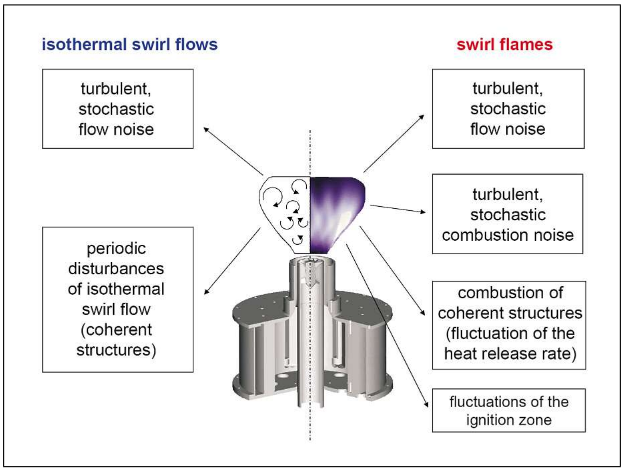

Figure 14. Important sources of combustion noise.

The part of the total sound emission caused by the turbulent flow (isothermal or with reaction) is given by the turbulent swirl flow with its benefits e.g. fast, turbulent mixing or flame ignition stability and cannot be reduced significantly by design modifications. The additional noise, generated by reacting coherent structures (here: oscillation of the axial position of the inner recirculation zone), indeed, is avoidable without any loss of the above mentioned benefits of turbulent swirl flames.

The investigation of non-premixed flames showed the necessity of a stable and fixed in time ignition zone to prevent additional noise through fluctuations of the ignition point leading to an increase of the combustion noise of 3 dB.

A summary of the detected and separated main noise sources of swirl flows and flames is given in Figure 14. On the left hand side the sources of an isothermal swirl flow are summarized, on the right hand side the different sources for swirl flames for premixed and non-premixed conditions are given. Additionally, the enclosed configuration is up to 20 dB louder than the non-enclosed swirl flames including the different measurement positions, not dividing into direct combustion/flow noise and reflected at the combustion chambers wall sound waves.

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