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PASSIVE CONTROL OF COMBUSTION INDUCED NOISE IN AN AUXILIARY BUS HEATING SYSTEM

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Abstract

Combustion instabilities and combustion induced noise are currently a serious problem for industrial and domestic boilers, furnaces and auxiliary heating devices. Interaction of fluctuating heat release and the acoustic field may generate high amplitude pressure oscillations, which have a negative effect on the combustion process and represent a dominant noise source. Although the pressure amplitudes are usually not high enough to cause any structural wear, as in the case of gas turbine combustors, dominant low-frequency sound emission, associated with flame-acoustic interaction, is a source of customer annoyance. In this work, an auxiliary heating system with a thermal power of 30 kW was investigated from a thermoacoustic point of view. The objective was to reduce dominant low-frequency noise observed in the far-field. Acoustic measurements were used to determine the cause of those sound field components. It was found that the major noise contribution originated from a resonant quarter-wave mode interacting with the flame. A geometrical modification of the outlet geometry, reducing the reflection of acoustic waves, achieved a significant decrease in the low-frequency noise. As a result of a larger outlet velocity of the new configuration, however, higher frequency components, associated with the exhaust jet, were generated. To suppress the negative effect of higher jet noise without loosing the advantage of reduced combustion induced noise, the outlet geometry was further improved. The final design was shown to suppress the flame associated low-frequency noise and did not significantly increase the jet noise at higher frequencies, thus reducing the overall sound pressure level.

1. INTRODUCTION

One of the main issues for the design of low-emission aero-engines, stationary gas turbines, and industrial and domestic boilers and furnaces is the noise associated with enclosed flames [1].

Fluctuating heat release interacts with resonant acoustic modes of the combustion chamber and may lead to exceptionally high pressure oscillations. If certain phase relationships between the acoustic waves and the unsteady heat release prevail, the linearized thermoacoustic system exhibits unstable modes, which grow in amplitude until limited by nonlinear mechanisms. These nonlinear effects are usually associated with a saturation of the heat release response to perturbations in acoustic velocity. Already in case of a stable system, however, significant acoustic amplitudes can be observed. In this case, the independent source term of the flame drives the system close to the resonance frequencies [2].

Significant reduction of sound pressure levels can be obtained either by active or passive means. In the latter case, Helmholtz resonators [3, 4] and vortex generators connected to the burner [5] where successfully used to suppress thermoacoustic oscillations. Although active control methods often achieve a higher reduction in pulsation levels and, in addition to that, can be applied to varying operating conditions (see, e.g., Dowling & Morgans [6]), passive means are preferred by the industry. This is due to the fact, that a passive device is more reliable and has no impact on maintenance intervals (as, e.g., in case of a high-bandwidth fuel valve).

The combustion system considered here is an auxiliary heating device for busses that heats the cooling water circuit by burning diesel fuel. Different classes of thermal power are available for this device, however, the configuration that was investigated has 30 kW at design conditions. The burner is typically operated at equivalence ratios of $\phi = 0.6 - 0.75$. Diesel oil is injected through a spray nozzle and dispersed in a swirling flow. Figure 1 (right) illustrates the functional principle of the heating device.

The motivation for the investigation and the modifications, described in the following sections, was a distinct low-frequency sound component that was reported to be a source of customer annoyance.

2. EXPERIMENTAL SET-UP

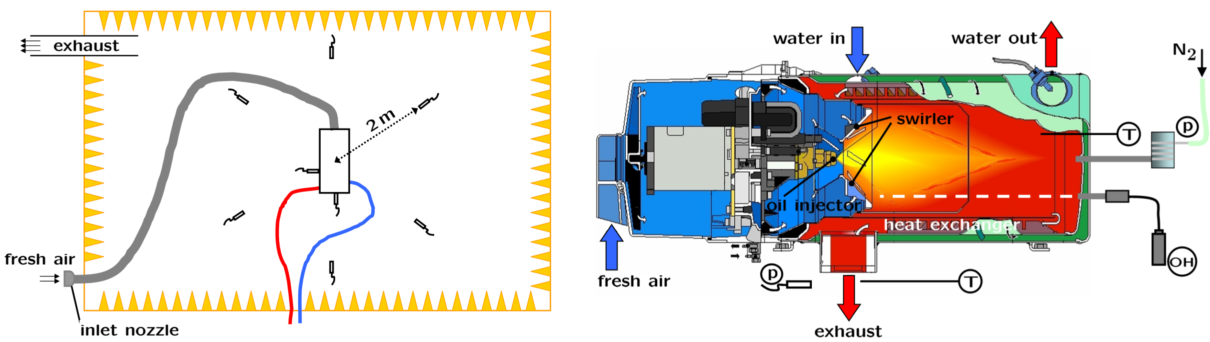

Figure 1. Experimental set-up in anechoic room with microphone locations (left) and detailed sketch of auxiliary heating device with sensor assembly (right)

The experimental investigations were performed in an anechoic chamber. Six microphones, uniformly spaced in angle, were placed in 2 m distance from the heating device (see Fig. 1, left). Although far-field conditions may not be achieved at low frequencies, the pressure signals recorded by these microphones will be referred to as such. A probe microphone, built upon the semi-infinite coil principle (see, e.g., Straub et al. [7]), was attached to the combustor back wall (see Fig. 1). The probe line was purged with a small amount of nitrogen to protect the microphone from the hot combustion gases. Another microphone was placed close to the exhaust exit plane. To detect oscillating heat release, a fiber optic cable was mounted close to the probe microphone. The cable was connected to a photomultiplier with an OH-bandpass filter. The combustion temperature was monitored with a thermocouple, also installed at the combustor back wall. A second thermocouple was placed in the exhaust stream leaving the exit plane. Figure 1 (right) shows a schematic of the heating device with measurement instrumentation.

Fresh air was taken from outside the anechoic room to prevent mixing with the exhaust. An inlet nozzle was used to determine the air mass flow. The oil nozzle was supplied a constant pressure so that the oil mass flow only depended on the oil temperature (mainly through viscosity). A cooler was used to keep the oil temperature constant. The oil mass flow was measured with a scale. Monitoring oil and air mass flows allowed to determine the equivalence ratio. However, due to the sensitive dependence of the combustion noise on the mixture ratio, an emission probe was additionally used to determine the fuel/air ratio based on the exhaust gas composition. The equivalence ratio could be varied by adjusting the air flow rate.

The heating device was mounted 1 m above the ground with the exhaust pointing downwards. All measurements were performed with a fully reflecting ground plate.

3. RESULTS

3.1. Baseline configuration

The objective of the investigation of the baseline configuration was to determine the source of the dominant low-frequency sound component observed in the far-field. Although the heating device offers several possible noise contributors (fan, oil pump, combustor, exhaust), a thermoacoustic source seemed likely. In order to prove the combustor to be responsible for the main noise emission, sound pressure recordings in the combustion chamber were correlated to those in the far-field. The sound pressure in the far-field was measured at six locations as shown in Fig. 1. As the directivity was not pronounced ($< 2\,\mathrm{dB}$), the results obtained from one of the far-field pressure sensors is considered to be representative, and only one spectrum is shown in the following.

Figure 2 displays normalized pressure spectra in the combustion chamber and in the far-field for equivalence ratios ranging from 0.54 to 0.77. Both spectra are dominated by low frequency sound components between 100 and 200 Hz. Combustor and far-field pressure are highly

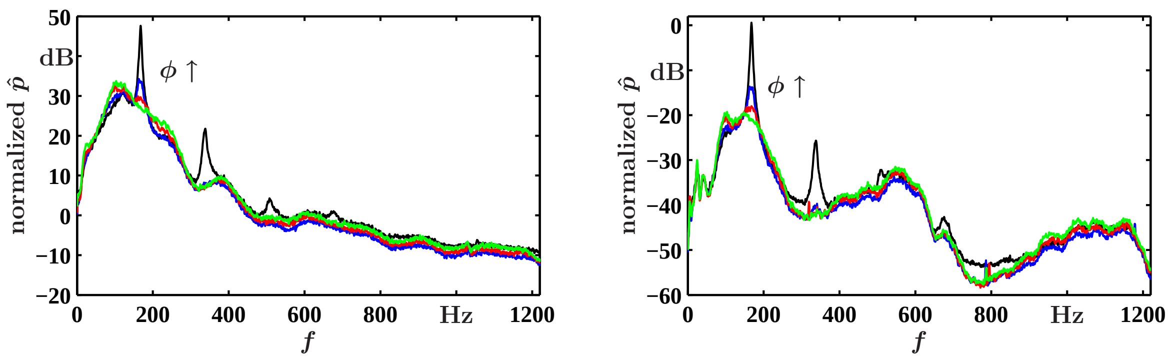

Figure 2. Normalized spectra of acoustic pressure in combustor (left) and far-field (right) for equivalence ratios of $0.54 - 0.77$

correlated in the low frequency regime (Fig. 3, left); this proves the combustor to be the origin of the dominant noise recognized in the far-field.

A decreasing mixture ratio slightly increases the whole sound pressure spectrum. This can be attributed to higher turbulence since a lower fuel/air ratio is achieved by raising the air mass flow (see Sec. 2). The effect is, however, small. For increasing equivalence ratio, a distinct spectral peak emerges around 165 Hz, corresponding to a thermoacoustic instability. At $\phi = 0.77$, the peak amplitude is almost 20 dB higher than the surrounding spectral band. Also, the harmonic becomes clearly visible, indicating nonlinear mechanisms. To confirm that thermoacoustic interaction is present, unsteady OH-chemiluminescence measurements were taken, as shown in Fig. 1 (right). At fuel richer conditions, the spectra of acoustic pressure and chemiluminescence in the combustor show distinct peaks at identical frequencies (Fig. 3, right). The chemiluminescence spectrum is rather noisy which is a result from the strongly limited optical access to the combustion zone (see Fig. 1, right). Nevertheless, the oscillation at the unstable frequency is clearly visible.

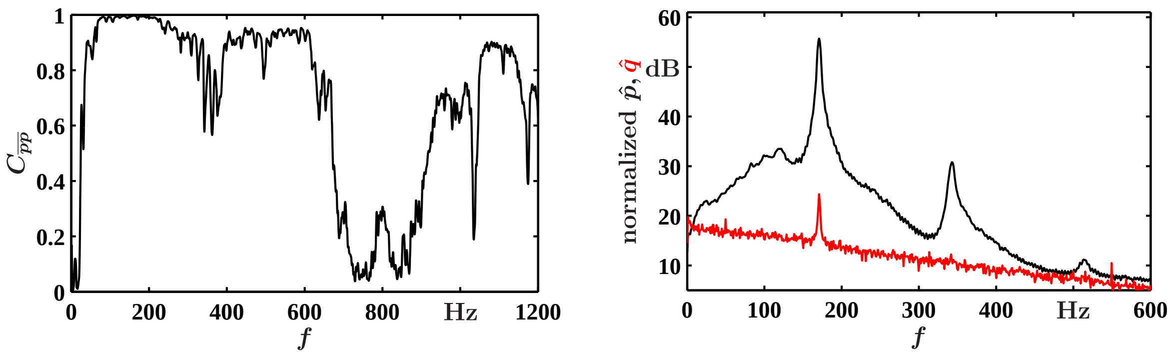

Figure 3. Left: Coherence between combustor and far-field pressure at $\phi = 0.7$. Right: Normalized spectra of acoustic pressure and OH-chemiluminescence at $\phi = 0.85$

Thermoacoustic oscillations in single burner configurations are often associated with a resonant quarter wave mode of the combustion chamber [8]. In case of the heating device considered here, however, the axial dimension seemed to be too small to host a quarter wave mode at frequencies clearly below 200 Hz and temperatures above 1200 °C. To understand which resonant acoustic modes are associated with the thermoacoustic oscillations observed, a finite element code was used to compute the eigenfrequencies of the combustor. The mode frequencies were determined from the Helmholtz equation

$$\nabla \cdot \left(\frac{1}{\rho_0} \nabla \hat{p} \right) + \frac{k^2}{\rho_0} \hat{p} = 0$$

subject to homogeneous boundary conditions. Here, $k = \omega / c$ is the wave number, and mean density ρ_0 and speed of sound c are functions of temperature. The mean flow is neglected. Since thermocouple measurements were taken only at two locations (combustor back wall and exhaust, see Fig. 1, right), no detailed information on the temperature field was available. For the FEM computation, a constant temperature of 1200 °C was assumed in the combustor and a linear decrease from 1200 °C to 400 °C was prescribed in the heat exchanger from combustor exit to exhaust. Due to symmetry, only one half of the combustor was modeled. Also, the fins in the heat exchanger (see Fig. 1, right) were not included in the FEM geometry.

A pressure node was prescribed at the exhaust exit plane; all other boundaries were assumed to be sound hard. The computational mesh and the pressure mode corresponding to the lowest eigenfrequency are shown in Fig. 4. The pressure in the combustor is almost constant. From heat exchanger entry to exhaust outlet plane, a quarter wave like shape can be recognized. The frequency of this quarter wave mode corresponds well with the experimentally observed resonance frequency. Therefore, it can be concluded that the dominant spectral contribution to the sound field observed in the far-field results from the interaction of the lowest acoustic mode with the unsteady heat release in the flame. All other eigenfrequencies that were found were significantly larger (Table 1). It should be noted, however, that due to the symmetry assumption, not all azimuthal modes were found.

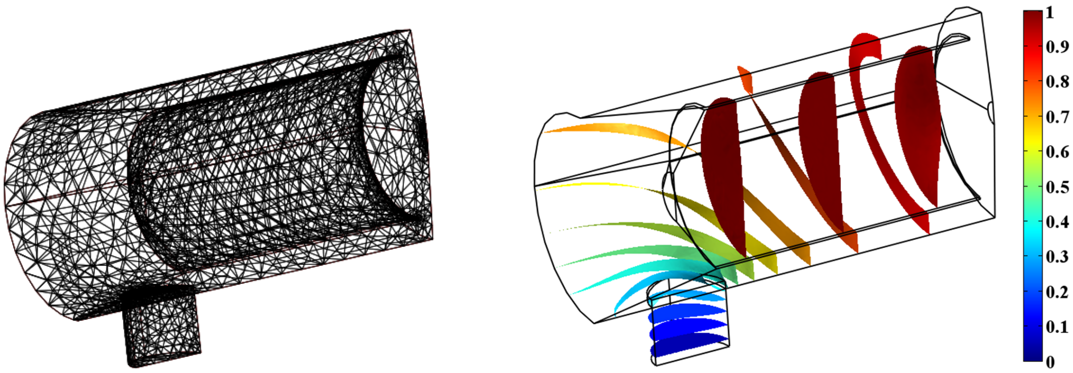

Figure 4. Computational mesh (left) and isosurfaces of the pressure distribution corresponding to the first resonant mode at 149 Hz (right)

The results presented in this section show that the dominant noise contribution in the low frequency regime, which is observed in the far-field, originates from the acoustic-heat release coupling in the combustor. For fuel/air ratios close to the stoichiometric one, the interaction generates sufficient acoustic energy that a thermoacoustic instability is established. Nevertheless, even in the stable case, the resonant acoustic mode driven by the flame is the major noise source.

Table 1. Lowest eigenfrequencies

n	1	2	3	4	5	6
f_n/Hz	149	658	1191	1375	1658	1834

3.2. Orifice in outlet nozzle

A crucial component in the thermoacoustic feedback loop is the reflection of acoustic waves at the combustor outlet [8]. Therefore, an efficient way to reduce flame-acoustic interaction is to decrease the reflectivity of the acoustic boundary condition at the exit plane. This strategy is rather unfeasible for real gas turbines, where the acoustic waves are reflected at the nearly choked turbine inlet. Nevertheless, in single burner test rigs it has been shown by active and passive means, that reducing the magnitude of the reflection coefficient at the combustor outlet (which is essentially increased damping) significantly lowers the thermoacoustically induced pressure oscillations [9, 10]. For low Helmholtz numbers, Bechert [11] gave a remarkably simple method to establish an anechoic termination of a flow duct. This is achieved by exploiting the mechanism of conversion of acoustic energy to fluctuating vorticity. If the duct exit is contracted in such a way that the Mach number of the flow in the contraction is equal to the area ratio, a highly absorbing boundary condition at the duct exit is obtained [11]. Paschereit et al. [12] successfully applied this technique to a gas turbine burner test rig. To achieve anechoic outlet conditions in the auxiliary heating device considered here, an orifice with the proper contraction ratio was placed in the exhaust exit plane.

The coherence of the pressure signals in the combustor and at the exhaust clearly shows the effect of the outlet contraction (Fig. 5, left). In case of the open outlet of the baseline configuration, the pressure signals are fully correlated in the frequency range where thermoacoustic oscillations occur. With the proper area contraction at the outlet, the coherence is reduced significantly (Fig. 5, left). The lower reflectivity at the outlet has a strong effect on the pressure spectrum in the far-field (Fig. 5, right). The low-frequency hump corresponding to the thermoacoustically resonant quarter wave mode is fully suppressed by using the orifice in the exit plane. This results from the reduced thermoacoustic feedback, which is achieved through lower reflectivity of the outlet boundary condition.

It can be also noted, however, that there is a significant increase in higher frequency components around 1000 Hz. This has to be attributed to the larger jet velocity at the outlet. For the given jet-to-ambient temperature ratio, the frequency range of higher noise agrees well with the peak Strouhal number given by Tanna [13]. Although the jet associated noise is at a considerably lower level than that of the resonant quarter wave mode, the higher frequency components around 1000 Hz are especially undesirable since they represent the dominant contribution in the A-weighted sound pressure level. Orifices with a different contraction ratio were also studied. A smaller area ratio resulted in decreased jet noise at higher frequency components but also had a less significant effect on the suppression of the quarter wave mode.

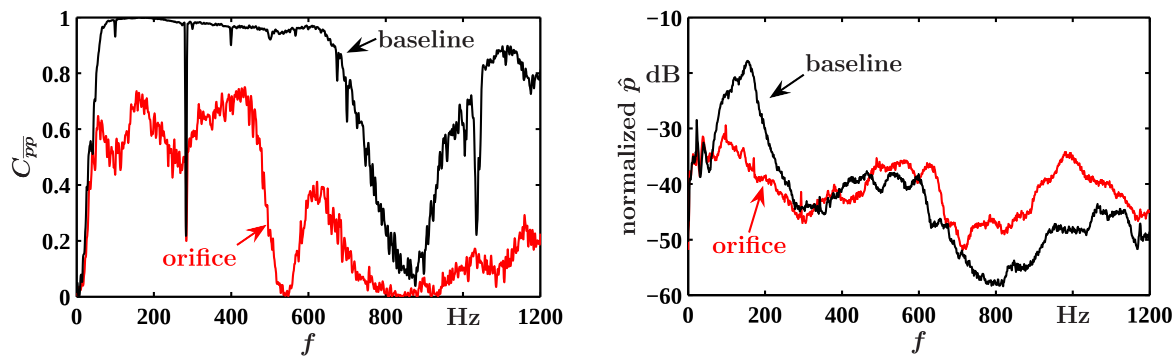

Figure 5. Left: Coherence of pressure in the combustor and at the outlet. Right: Normalized acoustic pressure spectra in the far-field. Results for baseline case and with orifice placed in the exhaust exit plane for $\phi = 0.65$

3.3. New nozzle design

The orifice in the outlet plane achieved the desired reduction in the low-frequency noise associated with the thermoacoustic feedback. The increased jet noise due to the area contraction was, however, not acceptable. For this reason, a solution had to be found which allowed for the area contraction – necessary for the suppression of thermoacoustic feedback – without generating excessive jet noise. A common method to control the jet noise emitted from aeroengine type nozzles makes use of so-called tabs or chevrons (see, e.g., Zaman et al. [14] or Callender et al. [15]). These are small geometrical objects that are placed at the nozzle lip and penetrate the flow. They have the effect to suppress the formation of large scale structures and enhance mixing with the ambient fluid through generation of axial vorticity [14]. In the present work, an orifice was combined with a short converging nozzle equipped with chevrons. The total area contraction was slightly less than that of the orifice described in the preceding section. Figure 6 shows the newly designed nozzle mounted in the outlet section of the combustor.

Figure 6. Newly designed nozzle in outlet section

The pressure spectra in the far-field at an equivalence ratio of 0.7 demonstrate the effect of the newly designed nozzle (Fig. 7). Compared to the baseline case, the low-frequency far-field pressure was significantly reduced. In addition, higher frequency components were not increased, as in the case of the pure area contraction. The thermoacoustic instability, which increased with increasing ϕ, was already apparent at $\phi = 0.7$, as demonstrated by the pressure peak at the quarter wave mode and its harmonic. This instability was completely suppressed by the new nozzle.

The variation of the total sound pressure level with equivalence ratio for the baseline case and the newly designed nozzle is shown in Fig. 7 (right). With the new nozzle, a significant reduction of the sound pressure level, for the whole range of equivalence ratios considered, is achieved. Compared to the baseline case, the reduction in total sound pressure level is at least 6 dB but can become more than 10 dB for larger fuel/air ratios. The increase of the sound pressure level at higher ϕ for the baseline case is due to the formation of a thermoacoustic instability, as mentioned above. Therefore, the reduction in SPL is higher for richer fuel/air conditions. When using the new nozzle, the SPL is almost constant over the ϕ-range measured.

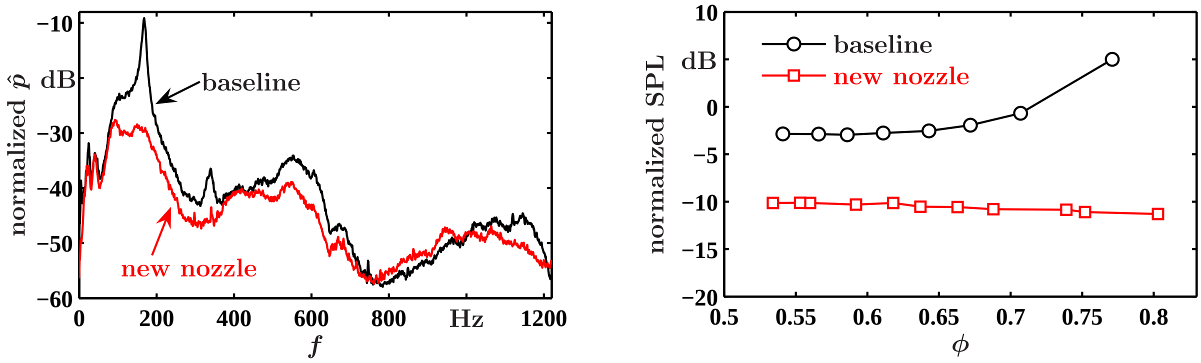

Figure 7. Left: Normalized acoustic pressure spectra in the far-field at $\phi = 0.7$. Right: Normalized sound pressure level vs. equivalence ratio. Results for baseline case and with new nozzle in outlet section

4. SUMMARY

An auxiliary heating device for public transportation busses was investigated with the goal to reduce combustion induced noise. The dominant low-frequency noise source was identified to arise due to flame-acoustic interaction. For fuel-rich operating conditions, a combustion instability, associated with the acoustic quarter wave mode, was found. An orifice placed in the outlet section was used to lower the acoustic reflectivity and thus decreased the thermoacoustic feedback. In this way, a significant reduction of the low-frequency part of the sound pressure spectrum in the far-field could be achieved. Due to the higher outlet velocity, however, the positive effect was accompanied by an undesirable increase in higher frequency components around 1000 Hz associated with the jet noise. The combination of an orifice and a chevron-like nozzle gave the best results, mitigating low frequency components without a noticeable increase in jet associated noise. In addition to that, thermoacoustic instabilities that were observed in the baseline configuration were fully suppressed with the newly designed nozzle.

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