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ACTIVE CONTROL OF COMBUSTION INSTABILITY USING SYMMETRIC AND ASYMMETRIC PREMIX FUEL MODULATION

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ABSTRACT

Active instability control was applied to an atmospheric swirl-stabilized premixed combustor using open loop and closed loop control schemes. Actuation was realised by two on-off valves allowing for symmetric and asymmetric modulation of the premix fuel flow while maintaining constant time averaged overall fuel mass flow. Pressure and heat release fluctuations in the combustor as well as NOx, CO and CO_2 emissions in the exhaust were recorded.

In the open loop circuit the heat release response of the flame was first investigated during stable combustion. For symmetric fuel modulation the dominant frequency in the heat release response was the modulation frequency, while for asymmetric modulation it was its first harmonic. In stable open loop control a reduction of NOx emissions due to fuel modulation of up to 19% was recorded.

In the closed loop mode phase-shift control was applied while triggering the valves at the dominant oscillation frequency as well as at its second subharmonic. Both, open and closed loop control schemes were able to successfully control a low-frequency combustion instability, while showing only a small increase in NOx emissions compared to, for example, secondary fuel modulation. Using premixed open loop fuel modulation, attenuation was best when modulating the fuel at frequencies different from the dominant instability frequency and its subharmonic. The performance of asymmetric fuel modulation was generally slightly better than for symmetric modulation in terms of suppression levels as well as emissions. Suppression of the instability's pressure rms level of up to 15.7 dB was recorded.

(Keywords: active instability control, phase-shift control, fuel modulation, combustion)

NOMENCLATURE

Φ fuel equivalence ratio
MFF ratio of modulated fuel flow to overall fuel flow (modulated fuel fraction)
f_{valve} valve operation frequency
p' pressure fluctuation
u' velocity fluctuation

INTRODUCTION

Modern gas turbine technology relies on lean premixed combustion to satisfy stringent governmental emission restrictions. Premixing the fuel with large quantities of air before injecting both into the combustor significantly reduces the peak temperatures in the combustion zone and thereby leads to lower NOx emissions.

However, combustion systems operating in the lean premixed mode are highly susceptible to the excitation of high amplitude pressure fluctuations called thermoacoustic instability (Poinsot et al. 1987 [1], Candel 1992 [2]). These self-excited oscillations are a result of the interaction between unsteady heat release in the flame and the combustion chamber's acoustic field. The main consequences of thermoacoustic instabilities are increased noise, reduced system performance and reduced system durability.

As described by Rayleigh's Criterion (1945 [3]), self-excitation of a combustion system occurs if the fluctuations in heat release from the combustion process are in phase with the pressure fluctuations. Although the real instability processes are somewhat more complex (due to combustor dynamics, fluid dynamics, chemical kinetics, transport processes, flame kinematics, heat transfer, etc.), Rayleigh's Criterion pinpoints how control of thermoacoustic instability can be achieved. A common approach is to induce heat release fluctuations, which are out of phase with the pressure fluctuations.

Passive techniques were developed to control the combustion characteristics by modifications of the fuel distribution pattern and changes in the combustor geometry (Schadow and Gutmark 1992 [4]). Active control systems, which have the potential to be more adaptable to variable operational conditions, decouple the physical processes that excite combustion instabilities, such as mixing, acoustics, and heat release. They acquire sensors, actuators, and feedback control algorithms that are used to drive the actuators and process output from the sensors. The actuators are used to modulate the air or fuel supply into the combustor. Dowling and Morgans (2005 [5]) completed an extensive review of active control.

Unsteady fuel injection was investigated as an actuation method for control systems that can potentially be adopted in full-scale power generation or propulsion applications. Langhorne et al. (1990 [6]) demonstrated active control of a 250 kW ducted flame. They obtained a 12 dB reduction in peak pressure oscillations by injecting 3% additional unsteady fuel upstream of the flame holder. An active control system with spatially distributed actuators was proposed by Fung et al. (1991 [7]). They developed a model that included modulated secondary fuel injection into the combustion chamber controlled by a PI system. An adaptive active control algorithm was developed by Billoud et al. (1991 [8]), who used a microphone and photomultiplier as sensors and a driver unit coupled to the fuel supply line to modulate the fuel injection. Active control of instabilities in rocket combustors by modulating the heat release in conjunction with a real time identification controller was proposed by Neumeier and Zinn (1994 [9]). Menon (1991 [10]) used LES to show a reduction of 10 dB in the rms fluctuations level in a dump combustor with secondary fuel injection. Oscillated secondary injection of liquid fuel was used to control oscillations in ducted premixed flames (Sivasegaram et al. 1995 [11]). They obtained a 12–16 dB reduction in pressure oscillations, and showed that control by oscillating liquid fuel is sensitive to atomization and the location of injection. Closed loop active control of instabilities in a 500 kW dump combustor using modulated liquid fuel injection was reported by Yu et al. (1996 [12]). They emphasized the importance of the injection timing of the modulated fuel relative to the formation of air vortices during instability on the droplets dispersion. McManus et al. (1998 [13]) modulated the main fuel flow using a high speed solenoid valve in an open loop and closed loop control system aimed to study combustion instabilities in a simulated afterburner. Their experiments showed strong response of the combustion process to the fuel modulations. Secondary fuel modulation was also shown to be effective in reducing combustion instabilities in gas turbine combustors (Cohen et al. 1998 [14]).

Paschereit et al. (1999 [15, 16]) investigated active control methods to suppress combustion instabilities in a swirl stabilized burner. They incorporated symmetric and asymmetric fuel as well as equivalence ratio modulation into their open loop and closed loop control schemes and not only achieved instability attenuation but simultaneously reduced emissions.

In the present work, symmetric and asymmetric premix fuel modulation was applied to a generic ABB-type swirl stabilized burner by means of two on-off valves while maintaining constant time averaged fuel mass flow. This active control method was employed into an open loop and a closed loop control scheme to suppress thermoacoustic instabilities. The combustor's exhaust gas was also analysed to investigate the control method's impact on emissions.

EXPERIMENTAL SET-UP

Combustion Facility

All measurement results presented in this paper were obtained at the Institute of Fluid Dynamics and Engineering Acoustics (ISTA) combustion facility depicted in Fig. 1. The atmospheric low-emission combustor has a 500 mm long, water-cooled combustion chamber. To generate a thermoacoustic instability with a frequency close to those typically occurring in full-scale engines, a resonance tube (900 mm long) was attached to this combustion chamber.

The combustor incorporates a generic environmental burner (EV-10) designed by ABB with a cross-sectional area expansion ratio of 4 for flame stabilization. Fig. 2 shows a detailed sketch of the burner. It is composed of two half cones shifted in such a way that the air is forced to enter the cone circumferentially through two slots. The resulting swirling airflow generates a recirculation zone along the centerline at the burner outlet, thus stabilizing the flame in this region. In standard operation (i.e. without fuel modulation), the main (premix) fuel is injected through 62 boreholes, 0.7 mm in diameter each, which are distributed equidistantly along the burner's two air slots and fed from one common fuel supply. Mixing of swirling air and main fuel results in a nearly premixed combustion. Pilot fuel, here only used during start of the combustor, can be injected at the EV-10 cone apex using a pilot lance. For a detailed description of the burner see [17].

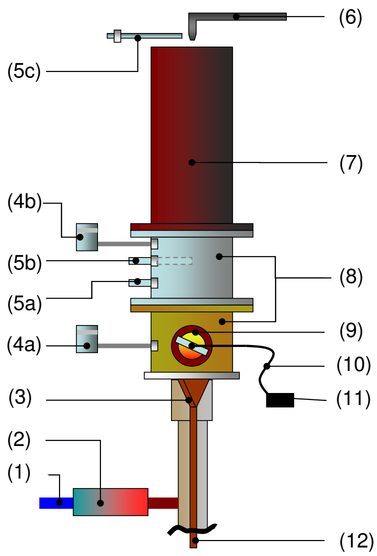

Figure 1. Test Rig Facility: 1) air supply, 2) preheater, 3) ABB EV-10 burner, 4a-b) condenser microphones, 5a-c) thermocouples, 6) emission probe, 7) resonance tube, 8) combustion chamber, 9) optical access to flame, 10) optical cable, 11) photomultiplier, 12) gas supply

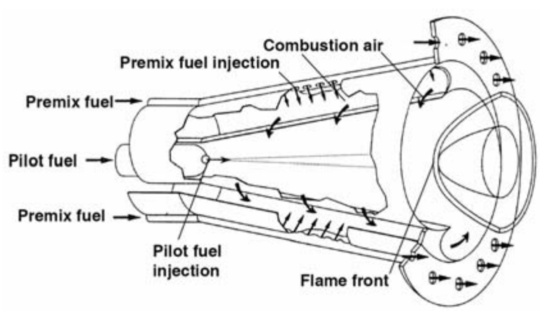

Figure 2. ABB EV-10 burner

Fuel Flow Control and Modulation

To allow for control of combustion instability, main fuel flow modulation was realised by employing two Bosch injection valves, commonly used in the automobile industry for the injection of gaseous fuel. The valves feature an on-off characteristic with a duty cycle period of 4 ms. A sketch of the fuel flow set-up is given in Fig. 3.

The overall main fuel flow was measured and controlled by a mass flow controller consisting of a coriolis flow meter and a slow-response proportional valve. Further downstream, the main fuel line was split, with the first end (black) feeding the burner's injection holes, and the second end (red) feeding the two on-off valves.

The on-off valves were further connected to two metallic tubes of 4.5 mm diameter by means of two 250 mm flexible

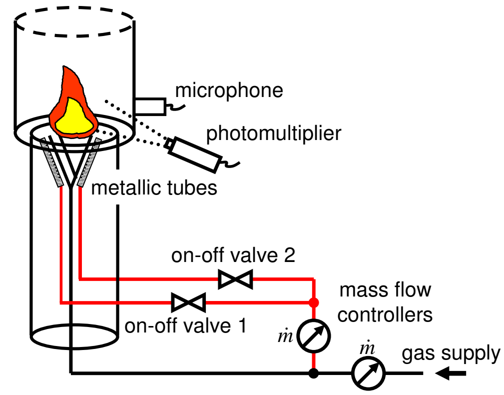

Figure 3. Fuel modulation set-up

Viton tubes joined to the combustion test rig upstream of the swirl burner. The metallic tubes were then attached to the burner 10 mm upstream of the two burner air slots, without blocking the air flow. Like the burner, the metallic tubes had equidistantly distributed boreholes for fuel injection into the premix air stream. Note that in contrast to the burner, each metallic tube had only 15 boreholes. However, the hole diameter was adjusted to achieve the same effective cross-sectional area as the burner boreholes.

For control of the modulated fuel fraction flowing through the on-off valves, a second mass flow controller was employed. The Bosch valves were controlled individually, thus allowing for symmetric and asymmetric modulation of a certain portion of the fuel flow.

Sensors

Pressure fluctuations in the combustion chamber were measured using two Brüel and Kjær 1/4-in. condenser microphones. Because of the harsh temperature conditions, the microphones were assembled in probe holders developed at the DLR Berlin, which operated like semi-infinite tubes to suppress reflections of the acoustic waves. The probe holders were continuously purged with Nitrogen to avoid water condensation and to provide for cooling. One probe was installed at the flame position, the other one further downstream (see Fig. 1). In the following, only the data recorded with the microphone at the flame position is presented, as higher amplitudes were measured at this position.

The heat release fluctuations were measured using a photomultiplier equipped with a narrow band-pass filter centered at 308 nm. At this wavelength, the photomultiplier captured light from OH chemiluminescence which is proportional to the heat release [18].

Both microphone and photomultiplier signals were amplified and low-pass filtered at 1 kHz to avoid aliasing. In closed loop control, the pressure fluctuations recorded by the microphone at the flame served as input to the controller. The descriptions of the open and closed loop control circuits, which were implemented in Matlab/Simulink, follow in Section 3 and 4. The Simulink model ran on a DS1103 PPC Controller Board (dSPACE) and generated command signals for both Bosch valves.

Combustion emissions were recorded using an ABB Advanced Optima system. This system captured the exhaust gases of the combustor through an emission probe positioned in the center of the resonance tube's exit cross section. The concentrations of NO, NO_2, CO and CO_2 were measured. The emission samples taken from the center of the exit cross section were assumed to be representative for the whole cross-sectional area due to the mixing in the relatively long resonance tube. For several operating conditions described in this paper, this assumption was verified by traversing the emission probe across the resonance tube's exit cross section without recording any significant differences in emission values. The recorded emission data was at all times normalised with the emissions at baseline conditions (no actuation).

Temperature data was taken at the combustion chamber centerline and wall as well as downstream of the resonance tube using a type B and two type K thermocouples.

Operating Conditions

Different operating conditions were investigated for this work. First, the actuator was tested at stable combustion to explore the system's response to the fuel flow modulation. Next, operating points with a high instability level were chosen to investigate the capabilities of symmetric and asymmetric fuel modulation to dampen the combustion instability.

All combustion tests were conducted with an air mass flow of 200 kg/h entering the burner at 300 K. The specific operation parameters for the presented data are given in the following sections.

To check for changes in the combustor's operating behavior due to the slight difference in fuel injection locations, preliminary tests at various unstable operating conditions and with different modulated fuel fractions were performed without the on-off valves in the second fuel line. The results did not reveal any significant changes for any of the fixed operating conditions when different MMF values were applied (hence the results are not shown here). This was to be expected, as the convective time delay between fuel particles injected through the metallic tubes and through the burner's boreholes is very small compared to the timescales of the dominant oscillation frequencies at unstable combustion.

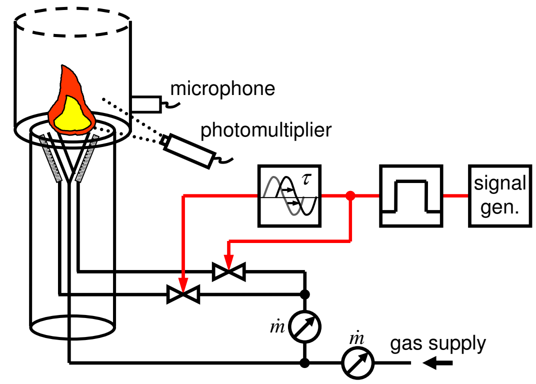

Figure 4. Open loop control scheme

OPEN LOOP CONTROL

The Control Scheme

Figure 4 presents the schematic of the open loop circuit used to achieve fuel modulation. A signal generator was employed to generate the trigger signal for the Bosch valves at the desired frequency. By introducing a time-delay function into the command signal of one valve, arbitrary phase-shifts between the two valves could be realised. However, only symmetric (τ = 0 ms) and asymmetric ($\tau = 1/(2f_{\text{valve}})$) operation of the two valves was investigated here. The maximum operation frequency of each valve was limited to 100 Hz.

The overall fuel mass flow and the modulated fuel fraction were also set in the control program and adjusted by the two mass flow controllers (control circuit not shown in sketch).

Actuator Response at Stable Combustion

Prior to testing the actuator in a full limit-cycling system, its ability to impact the thermoacoustic behavior of the combustor at stable operation was investigated. The test rig's resonance tube was removed, reducing the combustor length to approximately one third of the original size, and an overall equivalence ratio of 0.6 was chosen.

To make sure that both on-off valves had a similar effect on the combustor's heat release, they were operated individually with one valve closed at anyone time. This test was performed at a very low valve frequency of 20 Hz (to avoid interference between two consecutive cycles) and a MFF of 6 %. In Fig. 5 two periods of the phase-averaged heat release fluctuations of both valves are shown together with the corresponding valve command and pressure fluctuations. The results for the two valves are presented by solid and dashed lines, respectively. Note that the amplitudes of the pressure and heat release signals have been normalised.

The heat release fluctuations generated by the two Bosch valves match almost exactly, especially in terms of time delay

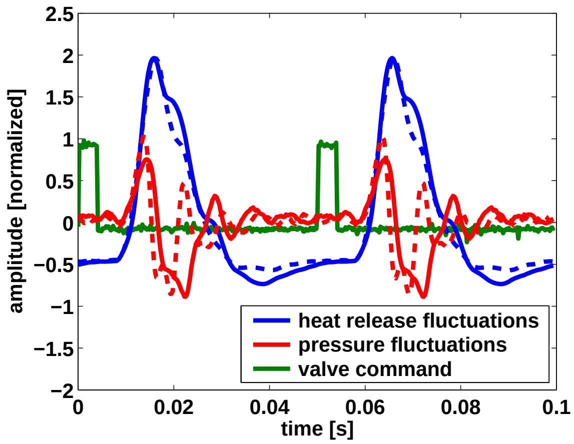

Figure 5. Comparison of release response to fuel modulation between the two Bosch valves. The results for the two different valves are plotted in solid and dashed lines, respectively.

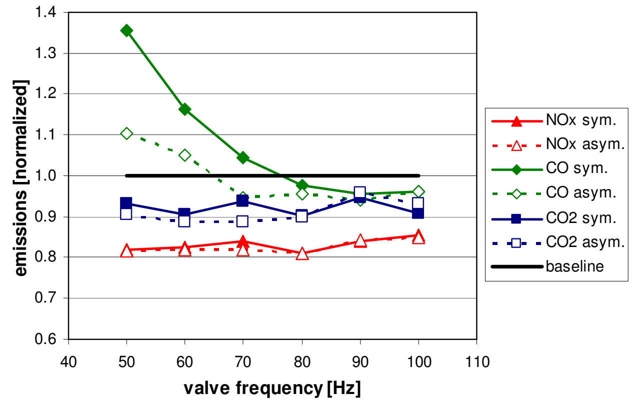

Figure 7. Emissions vs. valve frequency for open loop control (Φ = 0.6, MFF = 23%).

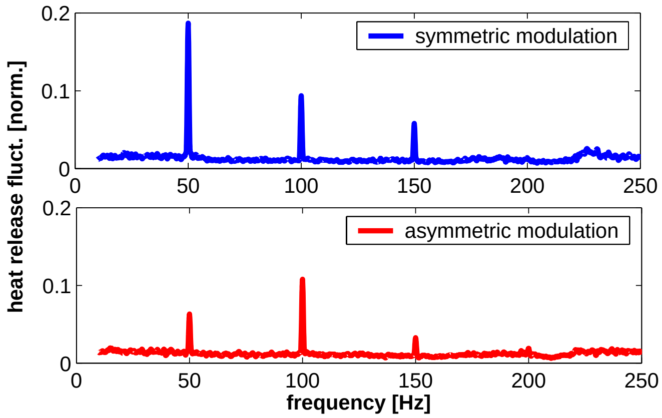

Figure 6. Effect of symmetric / asymmetric fuel modulation on the heat release spectrum.

between valve command and heat release response (about 9 ms), the width of the heat release signal (about 25 ms), its positive slope and maximum amplitude. Only in the decreasing part of the heat release signal can a small deviation be seen, most likely resulting from slightly different trajectories and hence convective time delays of some fuel particles before entering the flame region.

Note, however, that the heat release signal does not resemble square pulses as the valve command, but is rather smeared out. One reason for that was the distance between the valve and the fuel injection point (250 mm). In addition to that, the fuel was not injected directly into the premixed flame but into the burner slots (see Fig. 3), where it was subject to extensive mixing due to the swirling flows before reaching the flame.

The effect of changing from symmetric to asymmetric fuel modulation on the spectrum of heat release fluctuations is shown in Fig. 6. The Bosch valves were operated at 50 Hz modulating a fuel fraction of 23%. For symmetric fuel modulation the dominant oscillation frequency in the heat release signal is identical to the valve frequency, and higher harmonics have lower amplitudes. In the case of asymmetric fuel modulation the first harmonic of the valve frequency is dominant, indicating that actuation at twice the valve frequency is emphasised.

One probable cause of the higher harmonics obtained for symmetric fuel modulation is the on-off characteristic of the valves. Hot wire measurements, in which the velocity fluctuations of the fuel flow exiting the Bosch valves were recorded, confirmed the existence of higher harmonics in the corresponding velocity spectrum. However, detailed investigations on the fuel injection system exhibited a rather complex behaviour, that might also include, for example, the impact of the tubing between the valves and the injection points, the air-fuel mixing in the burner or non-linear flame response.

It was also found that fuel modulation with valve frequencies between 50 and 100 Hz led to a reduction of NOx- and CO_2 emissions with respect to baseline conditions of up to 19% and 11%, respectively (Fig. 7), with similar results for symmetric and asymmetric modulation. This can not simply be explained by a reduced temperature in the combustor as no such temperature drop was recorded by the thermocouples. For CO emissions an increase was observed for low valve frequencies ($f_{valve} \leq 70$ Hz). This increase was higher for symmetric than for asymmetric valve operation. For higher frequencies, however, this increase became smaller until even a slight reduction of CO emissions was achieved ($f_{valve} \geq 80$ Hz).

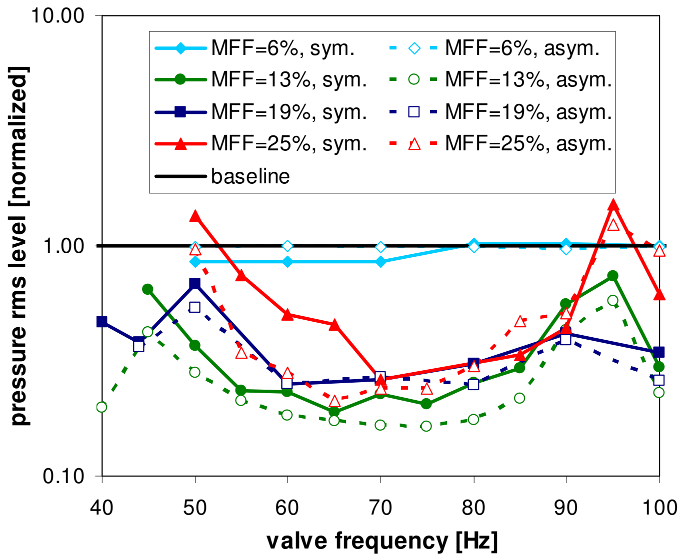

Figure 8. Normalized pressure rms vs. valve frequency for different MMFs and symmetric / asymmetric modulation.

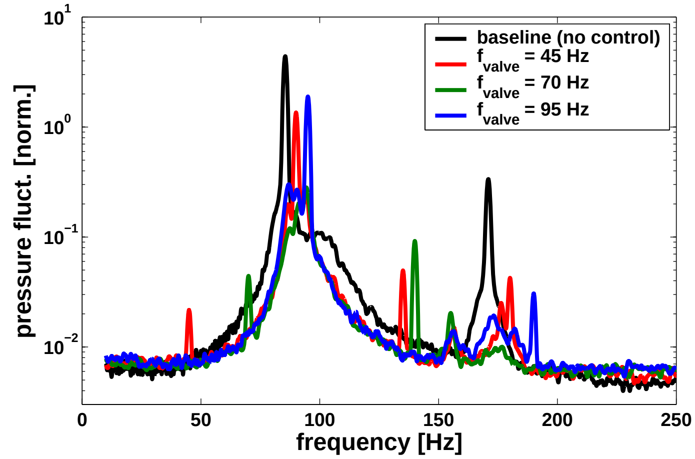

Figure 9. Spectrum of the pressure fluctuations measured for asymmetric fuel modulation with a MFF of 13%.

Instability Control

To test the capability of symmetric and asymmetric main fuel modulation to dampen thermoacoustic instability, the resonance tube was reattached to the combustor and an operation point with an equivalence ratio of 0.7 was chosen. At these baseline conditions, a strong instability with an oscillation frequency of 85.5 Hz was observed. Symmetric and asymmetric main fuel flow modulation was forced onto the combustion system with MFFs between 6 and 25% and valve frequencies from 50 to 100 Hz. Note that maintaining constant time averaged mass flow through the Bosch valves while, for example, doubling the valves' operation frequency had the effect of halving the fuel pulse per stroke.

From the obtained microphone data, the rms levels of the pressure fluctuations were calculated and normalized by the corresponding rms levels at baseline conditions. The results are presented in Fig. 8. Four main trends can be observed from the data:

1) Fuel flow modulation caused reduction of the pressure fluctuations at most of the operation points tested. A maximum dampening of 15.7 dB was achieved with asymmetric fuel modulation and a MFF of 13% at $f_{\text{valve}} = 75$ Hz.

2) An optimum MMF exists. Among the MMFs tested, this optimum was a MMF of 13%, where highest dampening was achieved. Higher MFFs resulted in less attenuation or even amplification of the pressure fluctuations. Lower MMFs also resulted in less attenuation and finally complete loss of control (values were identical to baseline case) especially for asymmetric fuel modulation or high valve frequencies, where the injected fuel mass per duty cycle of the actuator was small compared to symmetric injection at low frequencies.

3) In general, asymmetric fuel modulation proved to be slightly more effective than symmetric fuel modulation except for a MFF of 6%, where asymmetric injection and high valve frequencies resulted in loss of control. This superiority of asymmetric fuel modulation could be attributed to an interference with the periodic heat release of the symmetric mode.

4) Within the valve frequencies investigated, an optimum frequency range exists. Here, this optimum was found to be between 60 and 80 Hz, whereas frequencies around 50 and 95 Hz resulted in less efficient attenuation or even amplification of the pressure fluctuations. To examine in more detail the mechanism behind this trend, the spectra calculated from the measured pressure fluctuations at asymmetric fuel modulation with an MFF of 13% are plotted in Fig. 9. For comparison, the baseline pressure spectrum is also plotted.

At baseline conditions the dominant frequency of pressure oscillations was found to be 85.5 Hz with an amplitude of 55 dB above the noise level. Higher harmonics of this instability mode were found up to 513 Hz. However, as the instability mode itself and its first harmonic are the most dominant peaks in the pressure spectrum, the plot is limited to 250 Hz to allow for a more detailed presentation of the effect of fuel modulation. A second less pronounced instability mode was found in the pressure spectrum between 90 and 105 Hz with a comparatively small amplitude of around 23 dB above noise level resulting in a rather plateau-like shape within this frequency range.

Figure 9 shows the pressure spectra of the two cases with smallest attenuation ($f_{\text{valve}} = 45$ Hz and 95 Hz) and for a case with high attenuation ($f_{\text{valve}} = 70$ Hz). For asymmetric fuel modulation with valve frequencies or their harmonics within the frequency range of high baseline amplitudes, the response of the pressure signal to the modulation was quite strong with amplitudes of up to 47 dB above noise level for $f_{\text{valve}} = 95$ Hz.

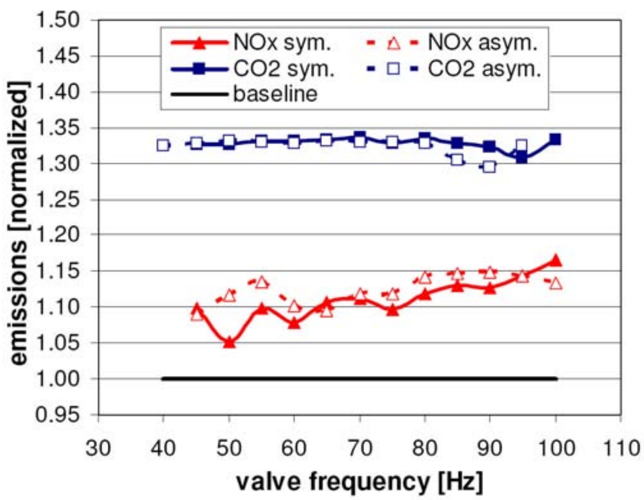

Figure 10. Emissions of NOx and CO_2 as a function of valve frequency.

Also, actuation at these frequencies resulted in entrainment of the dominant oscillation frequency, i.e. the dominant oscillation frequency was no longer the baseline instability mode, but the actuation frequency itself (or its higher harmonic for f_{valve} = 45 Hz). Additionally, the frequency range with amplitudes significantly higher above noise level was found to be much more narrow with fuel modulation than without. Furthermore, at around 171 Hz (the first harmonic frequency of dominant oscillation at baseline conditions) an amplitude reduction and a peak shifting toward the harmonics of the actuation frequencies were observed.

For valve frequencies outside the frequency range between 80 to 105 Hz, the amplitude within this range was further reduced (resulting in even higher attenuation) with the dominant peak still not matching the dominant peak at baseline conditions. At the same time the pressure response at the valve frequency and its harmonics became stronger. A similar effect was reported by McManus et al. [19].

The NOx and CO_2 emissions recorded during open loop instability control with a MFF of 13% can be seen in Fig. 10. CO emissions were below the measurement range of the corresponding sensor (concentrations < 0.5 ppm) indicating almost complete combustion and are therefore not shown here. Unlike in stable combustion, fuel modulation in unstable conditions resulted in an increase of 5 to 16% in NOx and 29 to 34% in CO_2. For CO_2 neither a difference between symmetric and asymmetric fuel modulation nor a significant change with valve frequency was observed. In the case of NOx a slight increase with higher actuation frequencies was noted with asymmetric modulation, generally leading to slightly higher NOx emissions.

A possible reason for the different emission trends due to fuel modulation at stable and unstable combustion, despite the different operation point, is enhanced further combustion down-

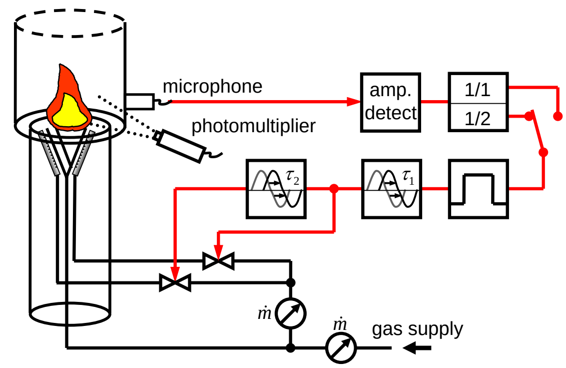

Figure 11. Closed loop control scheme

stream of the main flame due to the presence of the uncooled (and hence very hot) resonance tube. The very low CO emissions during operation with the resonance tube attached would support this argument.

Nonetheless, compared to the strong increase in NOx emissions caused by e.g. pulsed pilot fuel injection, a 16% increase in NOx emissions is still small. In an earlier investigation conducted on the same test rig, it was shown that pulsed pilot fuel indeed increased NOx emissions considerably [20] (i.e. increase of 200 to 500%).

CLOSED LOOP CONTROL

The Control Scheme

After considerable attenuation of a thermoacoustic instability was achieved by premix fuel modulation in an open loop control circuit, the same actuation mechanism was employed into a closed loop scheme, which allowed for phase-shift control. The set-up of this closed loop control scheme is shown in Fig. 11.

The unsteady pressure data acquired by the microphone at the flame position was fed back to generate the control signal for the Bosch valves. In principal, the OH signal could also be used as controller input, but it usually contains more noise. The reason for choosing the signal of the microphone at the flame position was that it detected a higher amplitude due to the quarter-wave mode shape.

In the feedback loop, the Matlab/Simulink program running on the dSPACE board detected every zero-crossing with positive slope (rising edge) of the input signal. A detailed description of this program is given in [21]. The Bosch valves were triggered in three different modes:

• For each detected rising edge the controller generated a trigger pulse. This pulse was phase-shifted and then passed to both on-off valves simultaneously. The valves were thus operating in symmetric mode at the fundamental frequency of the pressure oscillations.

• The controller generated a trigger pulse only for every second rising edge detected. In this case the valves were operated in symmetric mode at the first subharmonic.
• For each detected rising edge the controller generated a trigger pulse. Contrary to the first mode this pulse was forwarded to only one valve in turn, resulting in asymmetric valve operation at the second subharmonic.

The control board ran at a sampling frequency of 10 kHz. As for open loop control, the triggered valves modulated a certain fraction of the premix fuel resulting in a heat release fluctuation. The overall fuel mass flow and the MFF were set in the control program and adjusted by the two mass flow controllers.

The input signal to the controller was not band-pass filtered. Using a narrow band-pass filter on the input signal can help to track the dominant mode. However, such a filter generally induces a rapid phase change along the passband and adversely affects the control scheme if the frequency of oscillation is shifted by the controller. This mechanism can cause an intermittent loss of control [22].

Phase-Shift Control

The influence of time-delay on suppressing thermoacoustic instabilities was studied. The actuator was tested in the three different modes described previously: symmetric premix fuel pulses at the dominant instability mode as well as symmetric and asymmetric premix fuel pulses at its second subharmonic.

In Fig. 12 the normalised pressure rms level with respect to baseline conditions is plotted as a function of time-delay. The MMF was held fixed at 10%. The combustor was operated at $\Phi = 0.77$. For all tests (modulation at fundamental and subharmonics), the time-delay was first set to zero and then increased in steps of 0.8 ms to a maximum time-delay of 12 ms. A phase-shift of 360° with respect to the dominant oscillation frequency at baseline conditions corresponded to a time-delay of 11.7 ms.

Regarding the influence of time-delay, suppression performance of the actuator was best when operated asymmetrically at the 2nd subharmonic of the instability frequency with a time-delay of 4 to 4.8 ms. For these time-delays an attenuation of up to 9.4 dB was achieved. The lowest suppression for asymmetric fuel modulation occurred at 0 ms time-delay. However, the rms value of the pressure fluctuations was still below baseline conditions (1.7 dB).

In the case of symmetric fuel modulation at the 2nd subharmonic, the change of the pressure rms level with time-delay was similar to asymmetrical fuel modulation. The suppression was generally slightly lower, achieving a maximum attenuation of 9.3 dB at a time-delay of 4.0 ms and minimum suppression (1.4 dB) at a time-delay of 0.8 ms. Additionally, the time-delay interval of best attenuation was found to be somewhat more narrow compared to asymmetric fuel modulation.

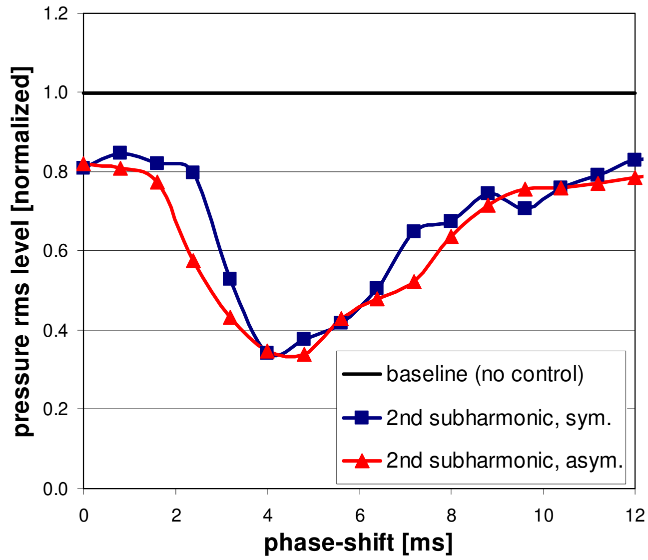

Figure 12. Normalized pressure rms level vs. time-delay (Φ = 0.77, MFF = 10%).

For fuel modulation at the fundamental instability frequency (and hence symmetric fuel modulation) preliminary tests indicated a similar behavior as for excitation at the 2nd subharmonic in terms of the time-delays corresponding to minimum and maximum suppression, while slightly lower maximum attenuation and slightly higher minimum attenuation was achieved compared to modulation at the 2nd subharmonic (symmetric as well as asymmetric).

Figure 13 shows the normalized NOx, CO and CO_2 emissions recorded during symmetric and asymmetric fuel modulation at the second subharmonic as a function of time-delay. As in open loop control, an increase in NOx and CO_2 with respect to baseline conditions (no control) was observed for closed loop control over the whole phase-shift range. Simultaneously the CO emissions decreased. The difference in emissions compared to baseline conditions was highest for time-delays within the range of high attenuation, i.e. around 4.8 ms (see Fig. 12). Here, a maximum increase of 46% in NOx emissions and 66% in CO_2 emissions was measured. This result highlights the quandary in the design of low NOx instability control strategies. The maximum decrease of CO was 67%, indicating almost complete combustion. The increase in NOx is still small compared to the increase caused by secondary pilot fuel modulation.

Compared to symmetric fuel modulation, asymmetric fuel modulation resulted in up to 21% lower CO emissions. Only minor differences were found for CO_2. In the case of NOx, asymmetric fuel modulation caused up to 9% lower emission levels at time-delays corresponding to high attenuation, whereas higher emission levels were recorded for time-delays corresponing to low attenuation (differences of up to 15%).

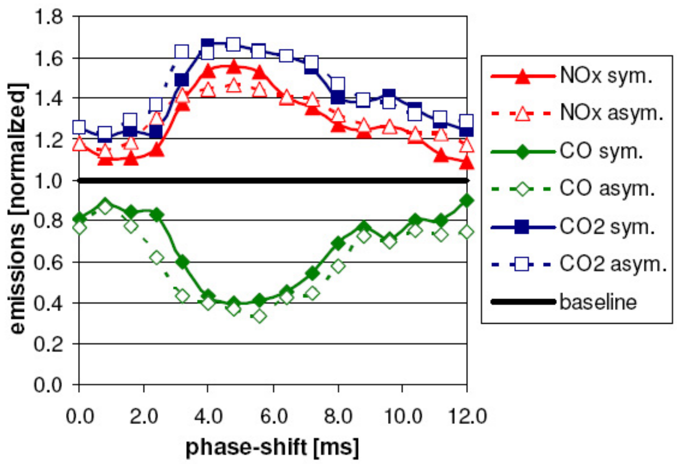

Figure 13. Normalized emissions vs. time-delay ($\Phi = 0.77$, MFF = 10%).

Figure 13 also pinpoints a common ambivalence between the attenuation of thermoacoustic instability and low emission combustion. For phase-shifts of 0 – 2 ms and 8 – 12 ms the attenuation levels due to fuel modulation are low, but so are the NOx and CO_2 emissions. However, phase-shifts of 2 – 8 ms result in much higher attenuation and lead to more complete combustion (indicated by the lower CO and higher CO_2emissions), but also cause an undesired increase of NOx and CO_2.

SUMMARY AND CONCLUSION

Active instability control was applied to an atmospheric swirl-stabilized premixed combustor using open loop and closed loop control schemes. Actuation was achieved by two on-off valves allowing for symmetric and asymmetric modulation of the premix fuel flow while maintaining constant time averaged overall fuel mass flow. Pressure and heat release fluctuations as well as NOx, CO and CO_2 emissions were recorded.

In the open loop circuit, the heat release response of the combustion system was first investigated during stable combustion. For symmetric fuel modulation, the dominant frequency in the heat release response was the modulation frequency itself, while for asymmetric modulation it was its first harmonic. In stable open loop control, a reduction of NOx emissions due to fuel modulation of up to 19% was recorded.

Modulation of the premix fuel flow by means of on-off valves with operation frequencies up to 100 Hz was shown to be effective in controlling a thermoacoustic instability with a dominant frequency of 85.5 Hz. In the closed loop mode phase-shift control was employed while triggering the valves at the dominant oscillation frequency as well as its second subharmonic. Maximum attenuation in the pressure rms level of 9.4 dB with respect to baseline conditions was achieved with a time-delay of 4.8 ms (= 148°) when operating the on-off valves asymmetrically at the second subharmonic of the dominant instability frequency. Simultaneously, only a small increase in NOx emissions was observed, compared to emission levels recorded for secondary pilot fuel modulation in tests with the same test rig.

In open loop control, best attenuation was recorded for a modulated fuel fraction of 13% at valve operation frequencies between 60 and 80 Hz, i.e. valve frequencies different from the instability and its subharmonics, with maximum attenuation in the pressure rms level of 15.7 dB when operating the valves in asymmetric mode at 75 Hz. As in the closed loop control case with maximum attenuation, a small increase in NOx was measured at this operation point.

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