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Helmholtz behavior and transfer function of an industrial fuel swirl burner used in heating systems

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Abstract

Combustion instabilities, due to dynamical phenomena in the combustion system, may lead to strong noise emissions in modern industrial and domestic heating devices. Lean premixed burners are often used in these systems helping to respect the pollutant legislations. On the other hand, these kind of burners are more sensitive to dynamical phenomena, and acoustical coupling with other system compounds may occur. In order to predict the acoustical response of a given combination of burner, boiler and chimney system, the acoustical transfer function of the burner is necessary. In this study we present the experimental determination of the transfer function of a 22 kW commercial fuel swirl burner, mounted in a corresponding boiler, for different operational conditions.

1. Introduction

Stronger regulations concerning the pollutant emission of industrial and domestic heating devices led to the development of a new generation of burners during the last few years. These burners are generally working with lean premixed flames, stabilized by recirculation zones. The dynamic response of such burners to acoustical excitation may be very strong, and coupling with other system elements may appear. In order to get a better control of the combustion, boiler manufacturers, which are not necessarily the same as the burner manufacturers, developed closed boilers, where the air supply is controlled by a ventilator. These behave like Helmholtz resonators with a capacity represented by the volume of the boiler and an inductance, corresponding to the chimney. The whole system, including burner, boiler and chimney, may oscillate continuously if an excitation source maintains this oscillation (Fig. 1). In particular, the burner itself may behave like an acoustic oscillation source, if it generates heat release oscillations at the resonance frequency and in phase with the pressure oscillations in the system (Rayleigh criterion) [1,2]:

$$\int p'(t)Q'(t)\,\mathrm{d}t > 0$$

This acoustical impact of heating systems is a growing field of interest for burner and boiler manufacturers [3], which aim to predict and prevent such coupling in their installations. In that context, the comprehension of these mechanisms and the determination of the burner response to an external excitation are of great interest.

Literature studies treating the dynamical behavior of industrial and domestic heating systems in the literature, are generally based on pure acoustical measurements, without simultaneous observation of the flame motion. Several authors have recently characterized the acoustic emission of heating systems, both, gas and light fuel oil burners, under different operational conditions [4–6]. Gonzalez and Asfaux [7] proposed a normalized acoustical measurement technique to characterize different boiler systems. Martin [3] gives some practical recommendations concerning the installation of domestic boilers in order to prevent sound emission and transmission to the environment. Kleine Jäger et al. [8] and Arribas [9] characterized and compared the acoustical impact of various domestic burners.

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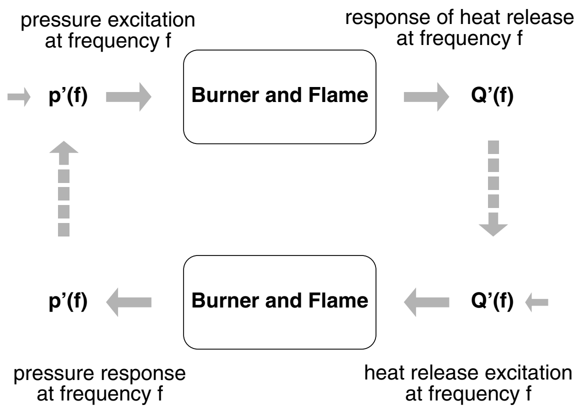

Fig. 1. Coupling mechanism between acoustic and heat release oscillations.

The fundamental sources of acoustical coupling, including studies about flame dynamics, has not been examined very often for the case of heating systems. A numerical study, concerning the response of a flat flame to acoustical excitation has been presented by Rook et al. [10]. They numerically determined the transfer function of the burner. In a previous study Zähringer et al. [11] established the relationship between acoustical emission and flame motion in a domestic gas burner.

In other combustion applications, an intense research activity is devoted to dynamical combustion phenomena (e.g. for gas turbines [12–14]) and to the theoretical and experimental determination of burner transfer functions [15,16] to get the response of the burner to a variation of pressure inside the boiler. Burner transfer functions generally have been determined experimentally in small model gas burners [17–20], although several authors underlined the importance of determining the transfer function also of real-scale burners [21–23], in order to model the complete system response. The main difficulty in real scale systems is to disconnect the pressure excitation from other combustion influencing parameters like stoichiometry or velocity. Another difficulty is the high "natural" sound pressure level of these burners, which necessities a very strong pressure excitation in order to get a sufficient signal-to-noise ratio.

In this study we determined the transfer function of a real scale, swirl-stabilized fuel burner in a domestic heating system submitted to pressure fluctuations in the furnace. Different equivalence ratios have been examined. The Helmholtz oscillator behavior of the system has been verified.

2. Experimental set-up

The experimental set-up is based upon a commercially available domestic heating system, composed of a Viessmann Vitola-Bifferral-RN boiler, equipped with an Elcko-Klöckner 22 kW swirl-stabilized burner. Optical access to the boiler is possible through a quartz window in the rear side (H in Fig. 2) of the boiler. This window,

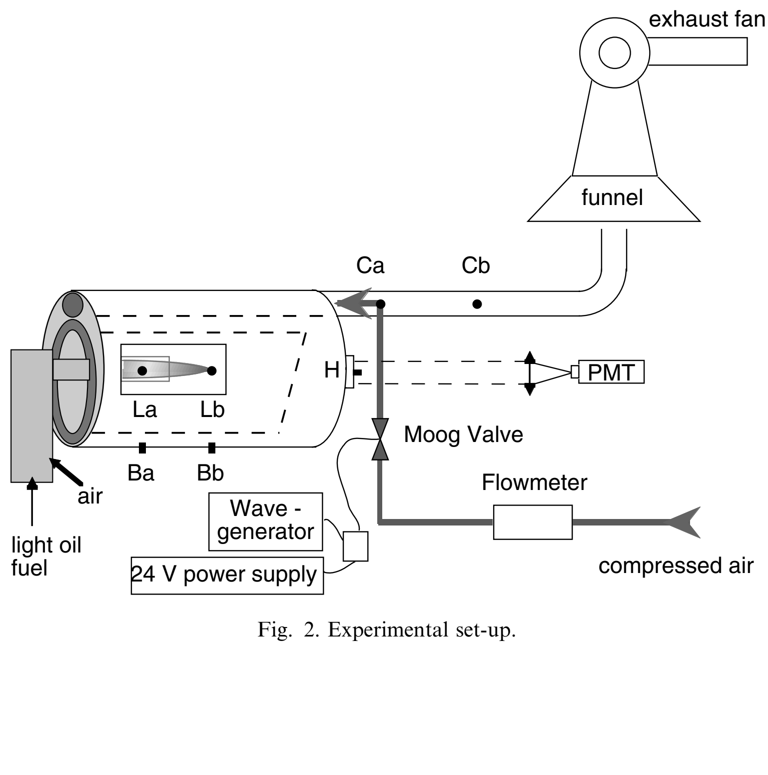

Fig. 2. Experimental set-up.

can be replaced by a steel plate equipped with a microphone port in order to enable acoustical measurements. Microphones can also be mounted on four other ports situated on the lateral and bottom side of the boiler (La, Lb; Ba and Bb). In the chimney one opening at 0.08 m from the boiler exit (Ca) has been used for the acoustical measurements, another opening could be used for gas analysis (Cb). The stainless steel chimney had a standard diameter of 125 mm and a standard length of 3.4 m, which has been varied for certain measurements. It ended in a funnel, decoupling the chimney from the exhaust gas extraction ventilator system.

For the acoustical measurements Brüel & Kjær 4136 microphones have been used. They are connected to the measuring ports by wave-guides at a distance of 10 cm to prevent them from heating. The wave-guide ends in a 25 m long tube, which eliminates reflected waves.

Analysis of the exhaust gas (CO, CO_2 and O_2) was used to control the operating point equivalence ratio of the burner and to keep it constant during the measurements. A portable Testo 342 analyzer has been used for these measurements. The overall equivalence ratio Φ for standard operational conditions was 0.512, with a domestic fuel consumption $\dot{m}_{\mathrm{fuel}}$ of 1.65 kg/h.

For the determination of the transfer function we used a microphone located on opening La (Fig. 2), in order to measure the pressure fluctuations inside the boiler. The heat release response of the flame is measured through OH chemiluminiscence. It is assumed [24,25] that the OH chemiluminiscence signal is representative for the integrated heat release in the burner. The whole flame emission is focused through window H by a lens ($f = 100$ mm) on the entrance pinhole of a photomultiplier (PMT). An interference filter ($\lambda = 313$ nm, $\Delta\lambda = 10$ nm) selects the OH-radical emission. The microphone and PMT signals are amplified, conditioned and recorded on a Brüel & Kjær 2148 spectrum analyzer. The acquisition frequency is 2048 Hz, leading to a bandpass of 0–400 Hz. The finally treated spectrum is the average of 1024 acquired spectra.

The main technical problem for the determination of the transfer function is to introduce controlled pressure oscillations in the boiler. They have to be strong enough to overcome the high "natural" sound pressure level of the boiler (here ≈90–115 dB depending on the frequency), in order to get a sufficient signal-to-noise ratio. The actuator system also has to introduce pressure fluctuations without simultaneously changing parameters like flame stoichiometry or air and fuel velocity at the burner inlet. It is also wanted to simulate as close as possible the acoustical coupling mechanism in the real system. This coupling mechanism, for domestic heating systems, is mostly determined by pressure oscillations on the downflow side of the flame. This corresponds to the upper branch of the thermo-acoustic coupling mechanism represented in Fig. 1, where heat release fluctuations are excited by initial pressure fluctuations. Consequently we decided to locate the actuator system on the downstream side of the flame, in the boiler or exhaust gas system.

Best results were obtained with a pulsed air jet, injected at the boiler exit as a counterflow to the exhaust gases (Fig. 2). The air jet is pulsed by a linear response, electromagnetic valve (MOOG D633) with a frequency response up to 200 Hz. A function generator delivered the corresponding frequency and amplitude signal. The mean pulsed air flow rate is low (≈9 m^3/h) compared to the exhaust gas flow (≈70 m^3/h). The mean velocity of the injected air is about 20 cm/s, compared to 1.6 m/s for the velocity of the exhaust gases. Due to this, an influence of the injected air on the equivalence ratio of the flame can be excluded, since furthermore the exhaust gases pass two times in the boiler cooling jacket between the flame and the pulsed air injection (about 1.5 m path length).

3. Helmholtz behavior of the examined boiler

For the analysis and modellization of the acoustical coupling in heating systems an assumption used to simplify the rather complex system description, is the hypothesis that the system is acting like a Helmholtz oscillator [4,5,10]. In this section the theoretical analysis and experimental verification of the Helmholtz-like behavior of the heating system examined in this study is presented.

3.1. Theoretical analysis

If a given boiler-chimney configuration can be regarded as a Helmholtz-oscillator, the general equation, describing this problem [26], can be written as

$$L\frac{\mathrm{d}^2}{\mathrm{d}t^2}(\partial\dot{m}) + R\frac{\mathrm{d}}{\mathrm{d}t}(\partial\dot{m}) + \frac{1}{C}\partial\dot{m} = 0 \qquad (1)$$

with the mass flow rate $\dot{m}$, the inductance L and the resistance R of the chimney and the capacity C of the boiler, where

$$L = \frac{\rho \cdot l}{A}, \quad R = \rho\frac{f \cdot u_0}{D \cdot A} \cdot L \quad \text{and} \quad C = \frac{V}{\gamma \cdot p_0}$$

with gas density ρ, the friction factor f, the mean velocity u_0 in the chimney, the mean static pressure p_0 and the isentropic coefficient γ.

Written in terms of velocity u in the chimney, Eq. (1) transforms to

$$L\frac{\mathrm{d}^2}{\mathrm{d}t^2}(\partial u) + R\frac{\mathrm{d}}{\mathrm{d}t}(\partial u) + \frac{1}{C}\partial u = 0 \qquad (2)$$

Table 1
Calculated resonance frequencies for the examined boiler (corrected with half the chimney diameter), for different temperatures and chimney length

l [m]	$T = 300$ K, $\rho = 1.177$ kg/m^3	$T = 450$ K, $\rho = 0.785$ kg/m^3	$T = 600$ K, $\rho = 0.588$ kg/m^3	$T = 1000$ K, $\rho = 0.352$ kg/m^3
3.4	18.0 Hz	22.0 Hz	25.4 Hz	32.9 Hz
1.4	27.7 Hz	33.9 Hz	39.1 Hz	50.6 Hz
0.94	33.4 Hz	40.9 Hz	47.3 Hz	61.1 Hz
0.49	45.1 Hz	55.2 Hz	63.8 Hz	82.4 Hz
0.08	89.3 Hz	109.4 Hz	126.4 Hz	163.4 Hz

For the configuration examined in this study the other parameters are:

Volume of the boiler	$V = 33.2 \times 10^{-3}$ m^3
Diameter of the chimney	$D = 0.125$ m
Cross-section of the chimney	$A = 12.3 \times 10^{-3}$ m^2
Length (maximum) of the chimney	$l = 3.4$ m

The resonance frequency ω_0 of the non-damped oscillator can be calculated by [26]

$$\omega_0 = \sqrt{\frac{1}{L \cdot C}} = \sqrt{\frac{A}{\rho \cdot l} \frac{\gamma \cdot p_0}{V}} \tag{3}$$

The resonance frequencies of our system for different mean temperatures T and different chimney length l can be calculated from these values (see Table 1).

As it can be seen from these values, where the lengths have been chosen as in the following experimental examinations, resonance of the system takes place in the low frequency range between 20 and 150 Hz. The resonance frequency increases with decreasing chimney length.

3.2. Measurement results

In order to verify, the assumption of a Helmholtz oscillator for the heating system examined here, experimental acoustical spectra have been recorded and several parameters have been changed in the system, like chimney length or microphone position.

A typical example of signals recorded inside the boiler at positions H, La and Ba and at Ca are represented in Fig. 3 for a 3.4 m long chimney. The boiler is operated under nominal conditions ($\Phi = 0.512$, $\dot{m}_{\mathrm{fuel}} = 1.65$ kg/h). It appears clearly that all three signals obtained inside the boiler are perfectly in phase and, at each instant, have nearly the same amplitude. The signal measured in the chimney is slightly delayed (about 4 ms), showing a small influence of the acoustics of the exhaust gas system. These results show that the pressure oscillations inside the boiler are in phase at every point

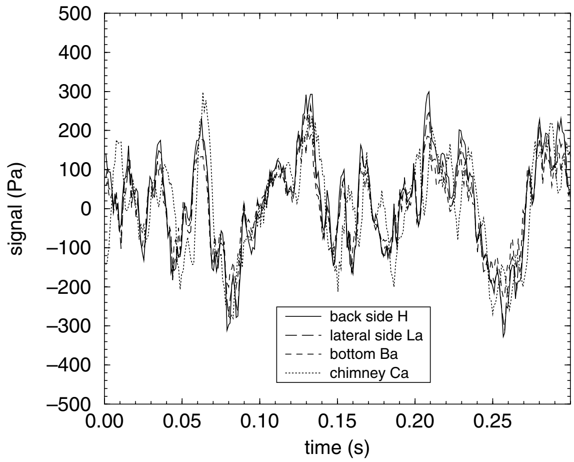

Fig. 3. Microphone signals from four different microphones located on the rear (H), lateral (La) and bottom (Ba) side of the boiler and at 8 cm from the boiler exit in the chimney (Ca) ($l = 3.4$ m).

and the boiler oscillates as a whole volume. It therefore can be considered as a Helmholtz resonator.

Typical acoustical spectra obtained under nominal operation conditions on the lateral side (La) and at the exit of the boiler (Ca) are shown in Fig. 4. The maximum sound energy is located at low frequencies. In the boiler (Fig. 4, top), two peak frequencies are located at about 20 and 45 Hz. They are rather large and thus typical for volume oscillations. These peaks can also be found in the chimney spectrum (Fig. 4, bottom), as well as a third characteristic frequency at about 90 Hz.

The variation of the chimney length is very important in order to explain the volume pressure oscillations inside the boiler. In Fig. 5 the effect of different chimney lengths on the spectrum measured in the chimney at Ca is shown. Five different lengths have been examined: 0.08, 0.49, 0.94, 1.4 and 3.4 m, the chimney diameter is kept constant at 125 mm.

The spectra in the chimney show very different peak frequencies for the five chimney lengths (Fig. 5). For the 0.08 m chimney a net decrease of the noise amplitude can be recognized. The first peak is situated at 24 Hz, the

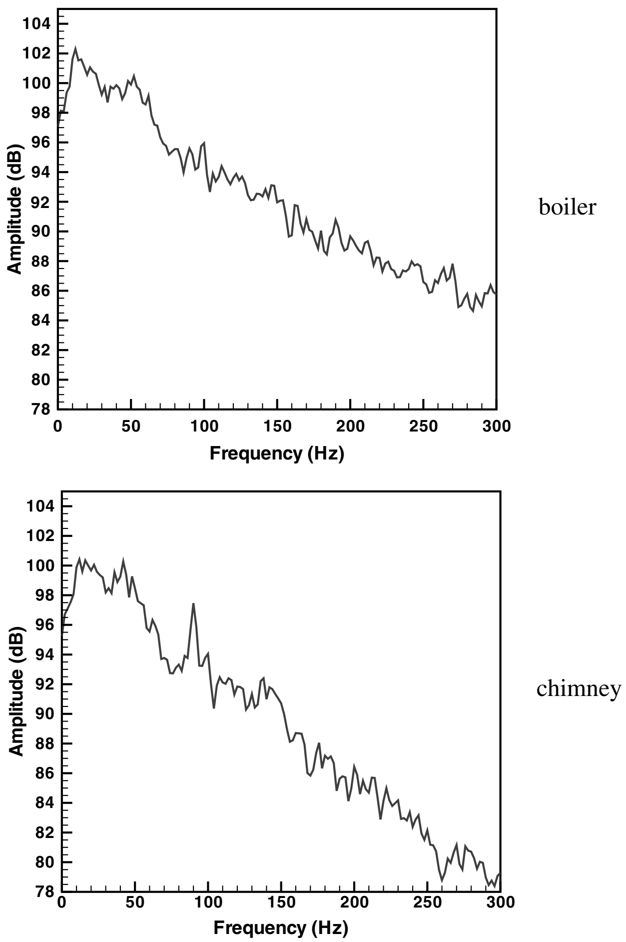

Fig. 4. Acoustical spectra at the lateral side (La) of the boiler (top) and at 8 cm from the boiler exit (Ca) in the chimney (bottom) ($l = 3.4$ m).

second one at 174 Hz. When increasing the chimney length, the first peak remains in the same zone between 15 and 25 Hz, but the second one is moving to lower frequencies: 140 Hz for $l = 0.49$ m, 104 Hz for $l = 0.94$ m, 82 Hz for $l = 1.4$ m and 88 Hz for $l = 3.4$ m. From 1.4 m on, the second peak rests in the region of about 80–90 Hz. A new very strong peak is appearing at about 40 Hz, for chimney length equal or greater 0.94 m. There is no increase of the mean amplitude between 0.94 and 3.4 m chimney length, saturation may be attained in the system.

These second peaks, measured at Ca with the different chimney lengths, can be compared to the calculated resonance frequencies in Table 1 for 450 K, which is the temperature of the gases in the chimney. The measured frequencies are always higher than the calculated ones. Thus Eq. (3) does not completely reflect the resonance influence of the chimney length. This may mainly be due to the complicated inner geometry in the cooling jacket of the boiler, which is situated between the combustion chamber and the chimney.

The displacement to lower frequencies of the peaks measured in the chimney and the saturation of the system from a certain chimney length on, validates the assumption of a Helmholtz oscillator. The mass of gas in the chimney stands for the inertia of the system and damping is growing with the chimney length and burnt gases mass involved.

These experimental results are summarized in Fig. 6. The boiler is oscillating as a whole volume, the chimney modifies this spectrum in function of its length for frequencies above 30 Hz. The strongest oscillation at around 20–30 Hz is independent of the system configuration, it is controlled by the burner "natural" noise. This part of the spectrum is probably related to the swirl frequency of the burner, which has been shown by flame visualizations. It can only be affected by a change of the burner design. The upper part of the spectrum is dominated by the Helmholtz resonator behavior, involving the characteristic of the boiler volume and chimney length.

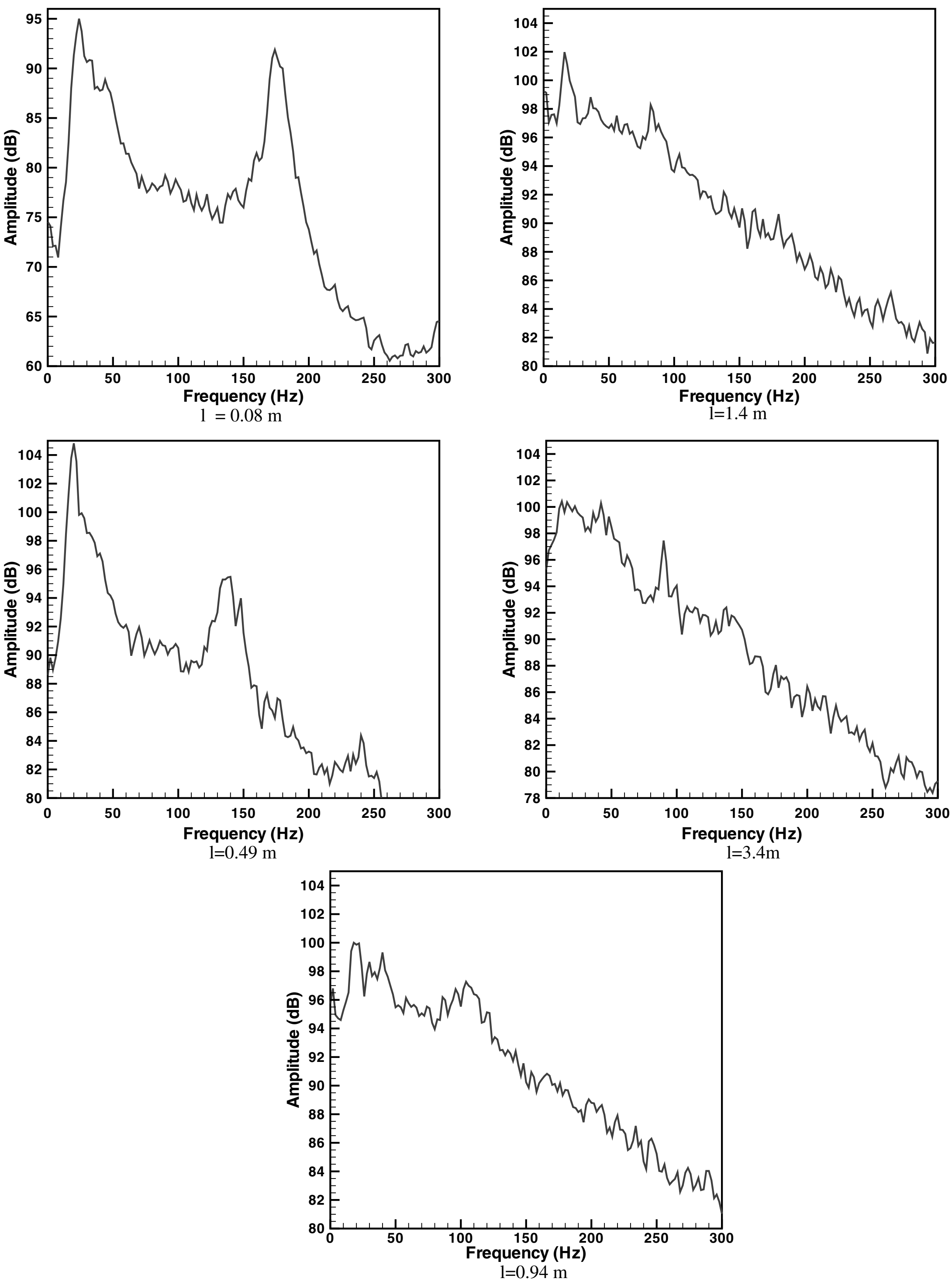

Fig. 5. Acoustical spectra at 8 cm from the boiler exit in the chimney at Ca for five different chimney lengths (graph has different scale for $l = 0.08$ m).

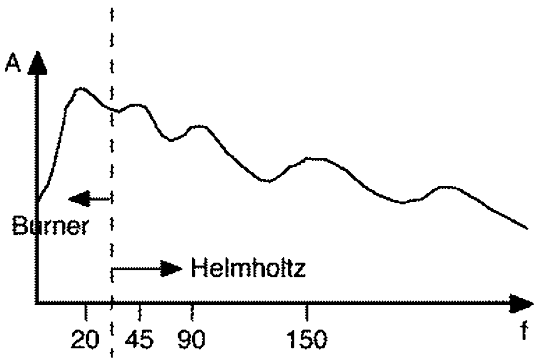

Fig. 6. Schematic representation of the examined system behavior.

4. Determination of the transfer function

In order to determine the transfer function of the system, a pulsed air jet pressure excitation is added as presented in Section 2 at the boiler outlet. The excitation frequency is varied in steps of 2–5 Hz from 10 Hz up to 130 Hz, the valve amplitude and the mean air flow rate (9 m^3/h) are kept constant. The obtained acoustical and OH-emission spectra are then cross-correlated to identify the transfer function. Since the system, as it is installed here, shows no particular acoustically excited mode, it is very suitable for the determination of the transfer function by an external pressure oscillation.

An example of simultaneously measured non-excited acoustical and OH-radical emission spectra is given in Fig. 7. The microphone was situated on the lateral side of the boiler (La). The three main peaks at about 20, 45 and 90 Hz, can be found in both, OH-emission and pressure fluctuations spectra.

Fig. 8 shows the externally excited acoustical signal at three different locations in the boiler obtained for an excitation frequency of 50 Hz. The signals in the boiler are perfectly in phase, thus showing that our actuator system produces pressure fluctuations in the whole volume of the boiler. Consequently the added pressure fluctuations will act on the whole flame, as in the Helmholtz oscillator behavior shown above.

A typical acoustical and simultaneous OH-emission spectra, obtained for an excitation at 50 Hz, is given in Fig. 9. The peak at the excitation frequency and its first harmonic are clearly visible in both spectra. This means that the flame responds to the external pressure excitation.

The transfer function, determined in that way for the nominal operation conditions of the boiler with an overall equivalence ratio of $\Phi = 0.512$ and a chimney length of 3.4 m, is represented in Fig. 10 for two different excitation air jet flow rates. The amplitude of the cross-spectrum between the acoustical spectrum and the OH-emission spectrum, increases with increasing

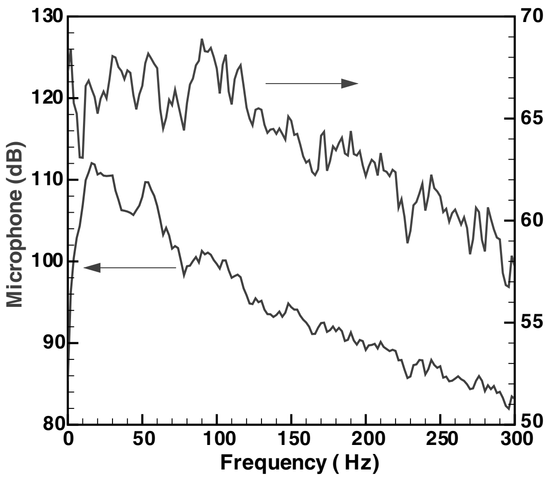

Fig. 7. Acoustical spectrum, measured on the lateral side (La) of the boiler at nominal conditions (lower curve, left axis), compared to the spectrum obtained from the OH-radical emission fluctuations measured with the PM (upper curve, right axis). Chimney length is 3.4 m.

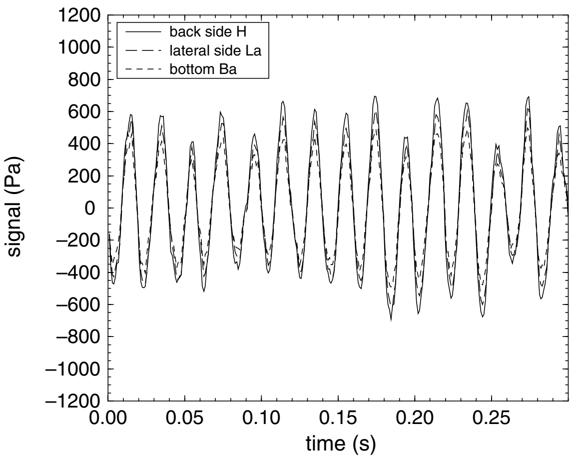

Fig. 8. Microphone signals from three different microphones located on the rear (H), lateral (La) and bottom (Ba) side of the boiler with an external excitation at 50 Hz.

air flow rate, since the injected pressure fluctuation is higher. The shape of the amplitude curves remains the same, independently of the flow rate. The phase of the cross-spectrum is nearly the same in both cases. In the following we chose to work with the higher air flow rate (9 m^3/h), in order to get a better signal-to-noise ratio.

It can be observed in Fig. 10, that the phase is almost zero at a frequency of about 90 Hz. Here the Rayleigh

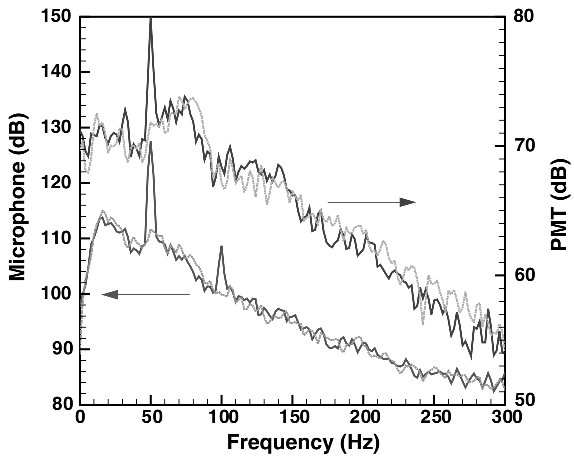

Fig. 9. Acoustical spectrum, measured on the lateral side (La) of the boiler (lower black curve, left axis), and spectrum of OH-emission fluctuations (upper black curve, right axis), both with an excitation at 50 Hz. For comparison a nominal non-excited spectrum is given for each curve (gray curves). Chimney length is 3.4 m.

criterion for the amplification of a pressure oscillation by the heat release of the flame is fulfilled. This explains the peak at 90 Hz in the acoustical spectrum (Fig. 7).

Fig. 11 shows the transfer functions for three different equivalence ratios ($\Phi = 0.665$, $\Phi = 0.512$ and $\Phi = 0.483$). They correspond to the upper and lower regulation limits of the burner air ventilator in the commercial system. The phase decreases with increasing equivalence ratio. For the higher equivalence ratio, the phase goes to zero at about 75 Hz. This means that the Rayleigh criterion now is fulfilled at this frequency and no longer at 90 Hz. In the case of the lower equivalence ratio the Rayleigh criterion is never fulfilled. The amplitudes of the cross-spectra do not vary much depending on the equivalence ratio. Amplitude tends to decrease for lower equivalence ratio.

The acoustical and OH-emission spectra for the higher equivalence ratio and without external excitation is represented in Fig. 12 for a chimney length of 2.15 m. With that chimney length the amplitude of an excitation at about 75 Hz should be amplified according to the results obtained before. Fig. 12 shows that this is the case for both, acoustical and OH-emission spectra. Around the frequency of 75 Hz, a peak appears in both spectra, which does not exist for the nominal equivalence ratio (Fig. 7). This peak corresponds to the zero phase observed in the transfer function (Fig. 11) at this frequency. The Rayleigh criterion leads for this frequency to an increase of noise emission around 75 Hz.

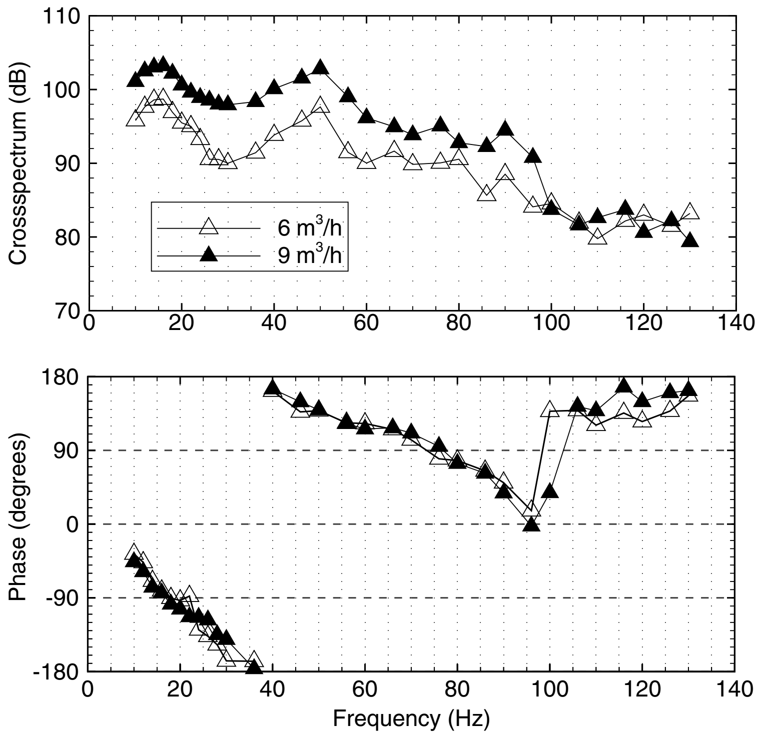

Fig. 10. Transfer function of the boiler at nominal conditions ($\Phi = 0.512$) for two different pulsed excitation air jet flow rates.

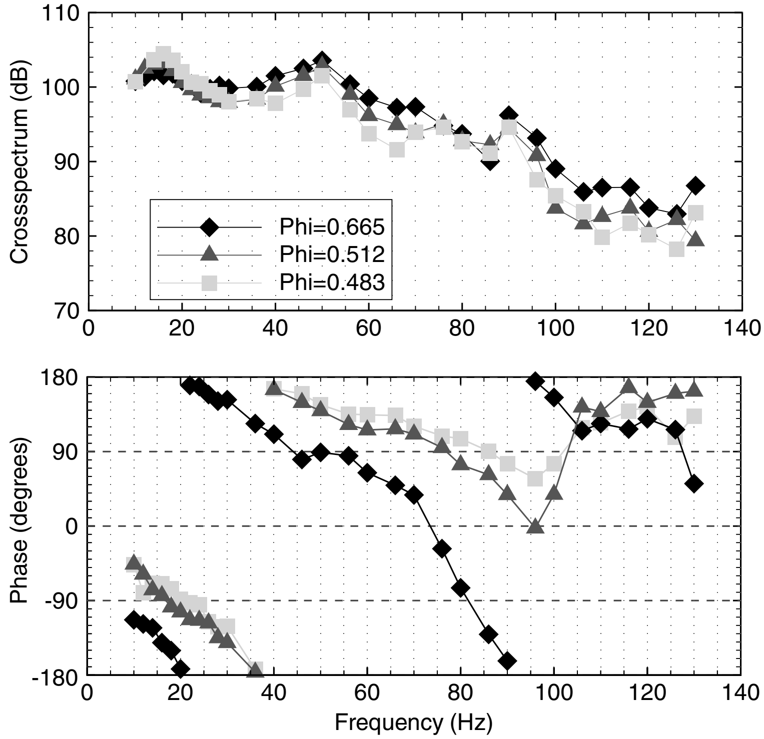

Fig. 11. Transfer functions for three different equivalence ratios. Pulsed compressed air injection flow rate is 9 m^3/h.

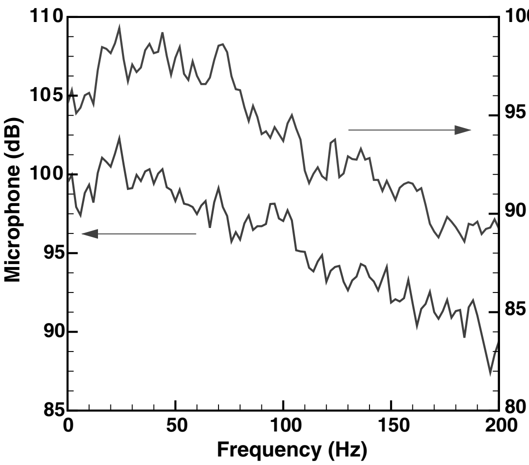

Fig. 12. Acoustical spectrum, measured on the lateral side (La) of the boiler (lower curve, left axis) and OH-emission spectrum (upper curve, right axis) at $\Phi = 0.665$. Chimney length is 2.15 m.

5. Conclusions

In this study, the thermo-acoustic coupling of a commercially available domestic heating system has been examined experimentally and its transfer function has been determined.

The acoustic measurements at different locations inside the boiler and chimney and for different chimney configurations, showed the Helmholtz-like character of the oscillations in the system.

An experimental set-up and method for the determination of the transfer function of practical heating systems have been developed and applied successfully. The equivalence ratio has a significant influence on the measured transfer function. Especially it determines at which frequency the Rayleigh criterion is fulfilled and specific coupling may appear in the system. In the case of the examined burner–boiler system this was the case at frequencies between 75 and 90 Hz.

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