Measurement and Modeling of the Acoustic Response in a High Pressure Combustor

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Abstract

In this paper, a one dimensional acoustic network model is presented which can be used as a design tool to predict the limit cycle pressure oscillations in a gas turbine combustor. Analytically represented models are combined with measured flame transfer functions and well defined boundary conditions. Additionally acoustic damping due to turbulence, acoustic reflection at contractions, modification of the acoustic speed of sound due to a mean flow and effect of temperature gradient that play a role in the acoustic modeling of combustion systems have been included in this network model. The model is applied on a high-pressure laboratory combustor. Finally, the measured and predicted dynamic behaviour in the combustor is compared. The results indicate the network modelling approach is a promising design tool for gas turbine combustion applications.

Introduction

Thermo-acoustic instabilities are likely to occur in many gas turbine combustors, when the thermal efficiency is optimized and the toxic flue gas emissions are minimized. To the designer and operator of the engine it is important to be able to predict how hazardous the instabilities are: will they develop to high amplitude limit cycle oscillations or will they be damped? The effect of the acoustic pressure oscillations on the structure and the consequences on the liner design play a crucial role with respect to the reliability of the gas turbine engine.

One dimensional acoustic network models are important tools to answer these questions. They are easy to apply and modify for acoustic systems. In the acoustic network model key elements to be introduced are the precise acoustic boundary conditions and the flame transfer function representing the flame behavior. These are very difficult to calculate from first principles due to complex interaction of the acoustic waves with the flame and geometry. For this purpose, measured flame transfer functions (FTF) and well defined reflection coefficients play an important role.

There are several ways to formulate an acoustic transfer matrix. The transfer matrix as used by Van der Eerden [1] relates the total pressure wave amplitudes to the mass flow perturbations. Hubbard and Dowling [2] used the forward and backward travelling pressure wave amplitudes instead of the total pressure wave amplitudes. Other authors, like Polifke [3] and Hobson [4] use a scattering matrix. This scattering matrix describes the transfer function between the pressure and velocity perturbation at both sides of the element. However the transfer matrix formulation used by Van der Eerden has several advantages. It uses total pressure wave amplitudes which can be measured directly using pressure transducers. This way, experimental results can easily be compared with the model. Furthermore, unknown transfer functions of any acoustic element can be measured in a direct way and included in the model.

The FTF is an important functional element to predict the acoustically unstable operating conditions of the combustor systems. The FTF describes the dynamic relationship between fluctuations in the inlet conditions of the flame and its acoustic response. The FTF has been studied in numerous papers for passive and active flames ([5] and [6] respectively). A method to measure the FTF with the use of pressure transducers and FTF calculations on basis of CFD results were discussed in the thesis of Van Kampen [7, 8]. Similar to Van Kampen's work Pozarlik [9] obtained the FTF with the use of experimental data and thermodynamic relations. Furthermore, the FTF is investigated for different geometries of the combustion chamber.

The experimental setup with its modular parts and data recording sensors is described first. Later the acoustic network model is presented, used to simulate acoustic characteristic of the setup. Details of the technique used to obtain the FTF are described. Finally the measured FTF and its modeled forms are implemented into the acoustic network model in order to simulate the flame. In the results section a detailed comparison of the measured and simulated pressure spectra at a chosen location for different operating points is presented and discussed.

Experimental Setup

The experimental combustion setup designed within the framework of the DESIRE project is able to work with a maximal thermal power equal to 500 kW at 5 bar absolute pressure. Natural gas is used as a fuel in premixed mode operation. The combustor consists of two coaxial square tubes. The inner square tube is the liner and the outer tube is the pressure casing. To decrease the overall temperature of the structural parts exposed to the flame and hot gases, a cooling air flow is introduced between the liner and pressurecasing. The cross-section of the combustion setup together with its modular parts and locations of data recording points is shown in Figure 1, where symbols 'P' and 'Ps' are dynamic and static pressure sensors respectively, 'T' are thermocouples and CCD stands for the camera for chemiluminescence measurements. The combustor mainly consists of three sections: the combustion, structural and cooling section.

The combustion section is located downstream the burner mouth. The burner opens into the inner square tube in which the combustion of the air/natural gas mixture takes place. To observe optically the flame the setup is equipped with a system of windows which are mounted in the liner and pressurecasing. Chemiluminescence and Planer Laser Induced Flourescence (PLIF) methods can be used to gather information about the in flame composition.

The Structural section is where the liner vibrations can be observed. At that location the liner has a reduced thickness of 1 (or optionally 1.5) mm, and 4 mm thickness is used for the other parts of the liner. The overall size of the setup is 150x150x1813 mm3.

In the cooling section the temperature of the exhaust gases is reduced by mixing first the hot flue gases with the cooling air from the cooling passage and then by a water spray. The secondary task of the cooling section is to provide a small pressure difference over the combustion chamber and cooling passage.

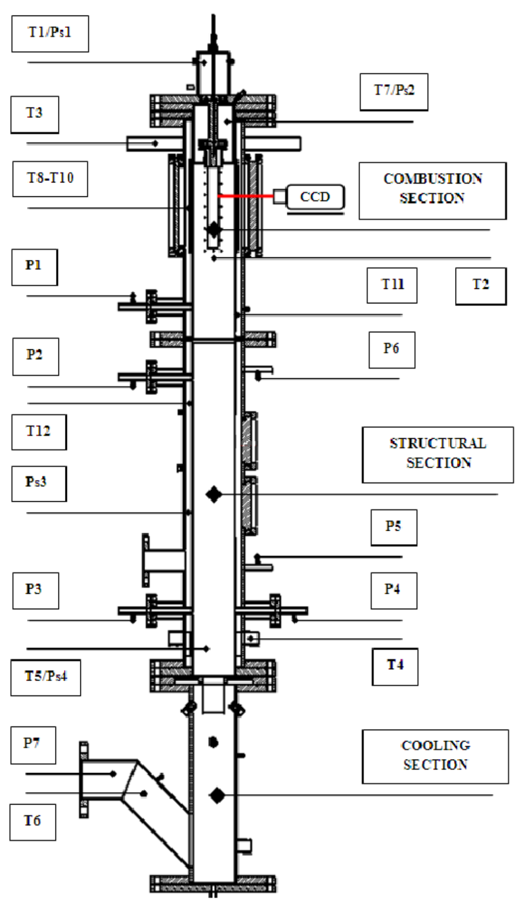

Figure 1: Experimental Setup

One-Dimensional Acoustic Network Model

At low frequencies the acoustic waves in combustion systems can be considered to be one-dimensional due to the high ratio between the acoustic wavelength and the typical cross-sectional dimensions of the combustion system [7]. As the name suggests, a 1D wave equation is used for describing the acoustic wave propagation in gases where viscous and thermal effects are neglected (wide tubes). The wave equation can be derived from the linearized forms of the mass momentum and equations, along with the linear relation between pressure and density for an ideal gas.

To predict the acoustic field in the system, the transfer matrix formulation is used. In the network model, the system is divided into elements which are represented by a transfer matrix. This matrix describes the relations between acoustic properties at both ends of an acoustic element. The most important effects, like acoustic damping due to turbulence, acoustic reflection at contractions, modification of the acoustic speed of sound due to a mean flow and effect of temperature gradient that play a role in the acoustic modeling of combustion systems have been included in the present network model. For the network model validation, an impedance tube measurement has been reconstructed with the network model, see Figure 2. Measurement and calculation have been done for 1 m length of tube, both ends closed, without mean flow and with speaker excitation. For further studies the measured and the modeled FTF have been implemented in the network model to simulate the flame effect.

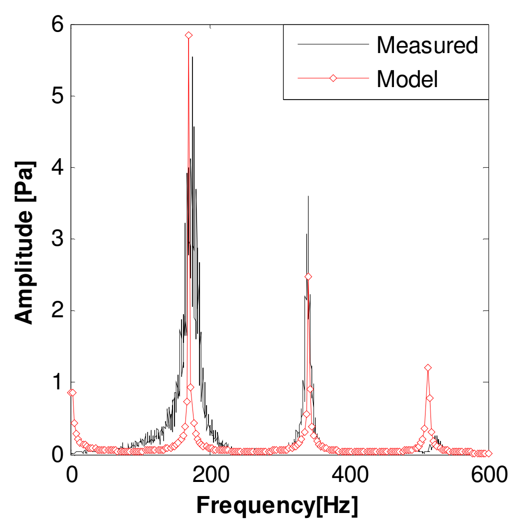

Figure 2: Validation of acoustic network model, impedance tube test

Flame Transfer Function

In this section the experimental method to measure the FTF is briefly presented. The FTF is extracted at elevated pressure (1.5 bar - 3.0 bar) for the operating points described in Table 1. The FTF is obtained entirely with the use of experimental data and thermodynamic relations. During a FTF measurement, the mass flow rate of the fuel $\dot{m}_f$ is perturbed by the MOOG valve with a piston that moves with a displacement δ'. This causes a heat release rate perturbation Q' in the flame. Since this perturbation is proportional to an acoustic sound source M', the perturbation can be measured by the pressure transducer in the combustion chamber. Since the instantaneous rate of heat release integrated over the combustor volume can not be measured directly, the FTF is divided on several, relatively simple to measure relations, see Eq. (1). An overview of the technique reconstructing the FTF is presented in Figure 3.

$$H_f = \frac{\bar{\dot{m}}_f}{\bar{\dot{Q}}} \cdot \frac{\dot{Q}'}{\dot{m}'_f} = \frac{\bar{\dot{m}}_f}{\bar{\dot{Q}}} \left[\frac{\dot{Q}'}{M'} \cdot \frac{M'}{p'_{meas}} \cdot \frac{p'_{meas}}{\delta'} \cdot \frac{\delta'}{\dot{m}'_f} \right] \quad (1)$$

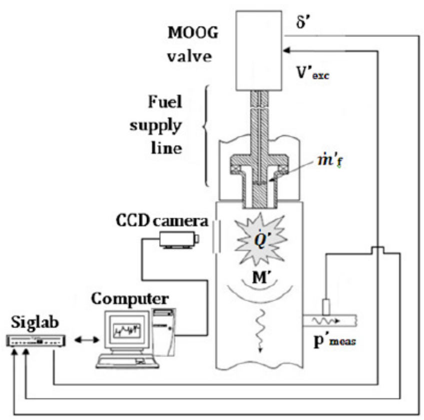

Figure 3: FTF measurement system overview.

The first factor in the bracket in Eq. (1) is known from thermodynamics as $\frac{\dot{Q}'}{M'} = \frac{c_0^2}{\gamma - 1}$. It is evaluated at adiabatic flame temperature and is assumed to be constant over the flame, as presented in [7].

The second factor i.e. $\frac{M'}{p'_{meas}}$ represents the transfer function between acoustic mass flow and local pressure perturbation. This function shows the acoustic behavior of the combustion system. Since the concentration of CH* radicals is linearly proportional to the heat release rate in the flame see [7], the spectral shape of the acoustic source is determined from chemiluminescence measurements of the light emitted by excited CH* radicals. For the spectral shape of pressure oscillations, the auto-spectrum of pressure transducer P1 is used.

The third factor $\frac{p'_{meas}}{\delta'}$ is obtained from the dynamic combustion experiment. The ratio between the cross-spectrum and auto-spectrum is taken to determine the transfer function between the pressure fluctuations measured by transducer P1 in the combustion chamber and the MOOG valve displacement.

The last factor $\frac{\delta'}{\dot{m}'_f}$ represents the ratio of the signal transmitted by the MOOG valve to the mass flow rate fluctuation of the fuel. This data is received from a separate experiment performed and described by Kleinlugtenbelt [10]. Finally a combination of all these factors in the FTF of the investigated combustion setup is presented in Figure 4.

The measurements are done for frequency fluctuations in a range from 40 Hz to 400 Hz with 5 Hz steps and amplitude of pulsation equal to 7.5% of the mean equivalence ratio. Two different operating points are shown in Table 1. For the frequency stepping a sweep-sinusoidal signal generated by a Siglab spectral analyser system is used. For CH* measurements a CCD camera is used with 1000 Hz sampling frequency which limits the CH* spectrum to 500 Hz.

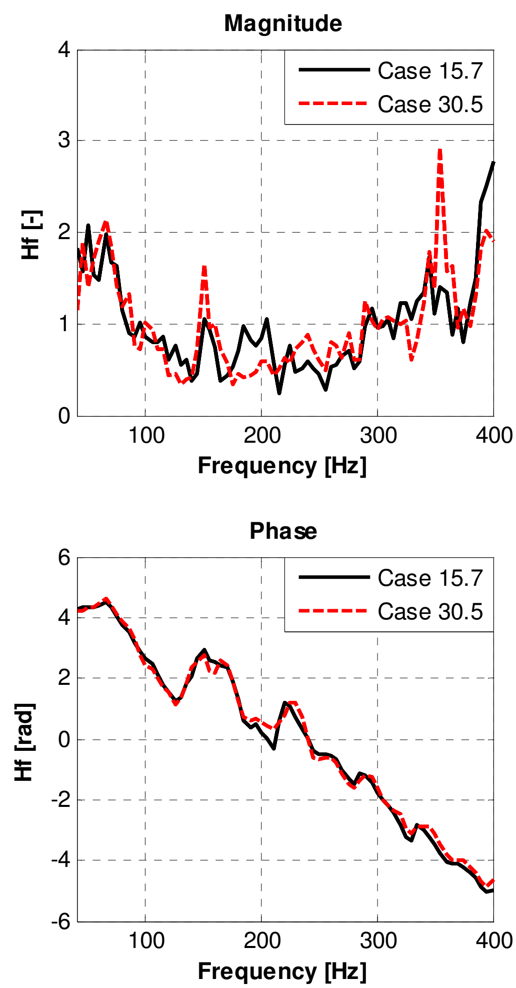

Figure 4: Flame transfer function of the Desire setup

The magnitude spectrum in Figure 4 shows slight differences between the different operating points. This behavior is in good agreement with the behavior of the transfer function reported in several references see [7, 11]. The phase spectrum shows a linear dependence between phase change and frequency and it is similar for both investigated cases. A constant convective time delay can be calculated for each spectrum by fitting the linear function. For operating point 15.7 the predicted time delay between the mass flow rate fluctuations and flame response is equal to 3.7 ms, whereas for the operating point 30.5 it is 3.6 ms.

In the FTF measurements, it is desired to have a constant amplitude of perturbation of the fuel mass flow at all considered frequencies. Due to a limitation of the MOOG valve the FTF can be measured up to 400 Hz (the valve has a cut-off frequency nearby 420 Hz). Above the cut-off frequency, the level of excitation decreases rapidly. Therefore an analytic model is used to extrapolate the FTF behavior for higher frequencies in the network model.

As an analytical fit to the FTF, the $n-\tau$ model is known in the literature [7]. In this model FTF, the H_f between the equivalence ratio perturbation ϕ' and the volume integrated heat release rate Q' was expressed as:

$$H_f = \frac{Q'}{\phi'} = n \cdot e^{-i\omega\tau} \tag{2}$$

where τ is the time delay between the excitation and the response, and n is an amplification factor.

As an addition to the $n-\tau$ model, the $n-\tau-\sigma$ model [12] is also studied. In this model, the heat release distribution has been approximated with a Gaussian function with a mean time delay τ and its standard deviation as σ. The $n-\tau-\sigma$ model can be written as:

$$H_f = n \cdot e^{-i\omega\tau} \cdot e^{-\frac{1}{2}\omega^2\sigma^2} \tag{3}$$

The flame is modeled in the network model by placing an acoustic volume source at the position of the burner mouth. As a validation, comparisons between the FTF and $n-\tau$ and $n-\tau-\sigma$ models have been made and shown in Figure 5 for the 15.7_2 case conditions. The experimental FTF is implemented into the network model and its pressure spectrum at the location of pressure transducer P1 is simulated from 40Hz to 400 Hz. Later the modeled FTF is implemented into the network model to see the differences. For this operating point, n is calculated as 17.81, the time delay τ is taken as 3.7 from phase spectra of the experimental FTF and its standard deviation σ is taken as $0.16 \cdot \tau$. Using the $n-\tau-\sigma$ model fit, a better approximation is achieved for the experimental FTF.

A comparison between the measured and simulated pressure spectrum at the location of pressure transducer P1 is discussed in the next section.

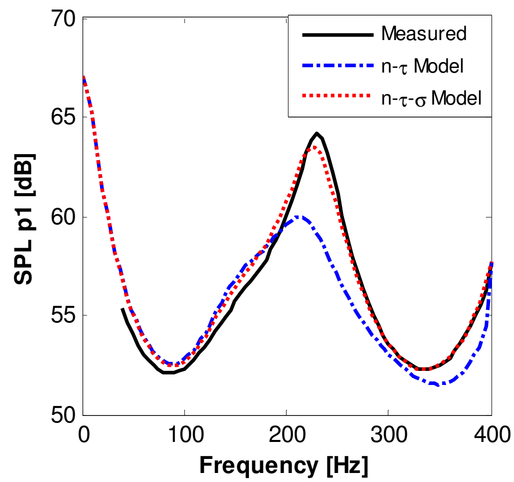

Figure 5: Measured and modeled FTF

Results and Discussions

In this section results are discussed for different operating points i.e., at different power and elevated pressure conditions (1.5 bar and 3 bar) as shown in Table 1. Date from operating points 15.7_1 and 30.5 were recorded during one experimental session, whereas operating point 15.7_2 was studied on another day. Changes in the outlet temperature profile can be explained by variations in the temperature of combustion air and the mass flow ratio between cooling air and combustion air (1-1.2). The difference in the preheated air temperature effects the adiabatic flame temperature and cooling air percentage which effects the temperature distribution along the liner and combustion outlet. The temperature gradient has a large effect on the acoustic eigenfrequencies.

Table 1: Investigated operation conditions.

Operating points	15.7_1	15.7_2	30.5
Thermal power [kW]	125	125	250
Absolute pressure [bar]	1.5	1.5	3.0
Adiabatic flame Temp. [K]	1834	1795	1834
Outlet Temp. [K]	1230	1100	1150

In the following figures the black curve corresponds to the experimental pressure spectra measured at the location of pressure transducer P1 and the curve with the circle marker is the result of the acoustic network model simulation for the same location of pressure transducer P1. In Figure 6, experimental data from operating point 15.7_1 is used for simulations. For this measurement the main instability is recorded at frequency of 450 Hz. whereas, the acoustic network model predicts it at

440 Hz. Secondary and higher order instabilities are also well predicted.

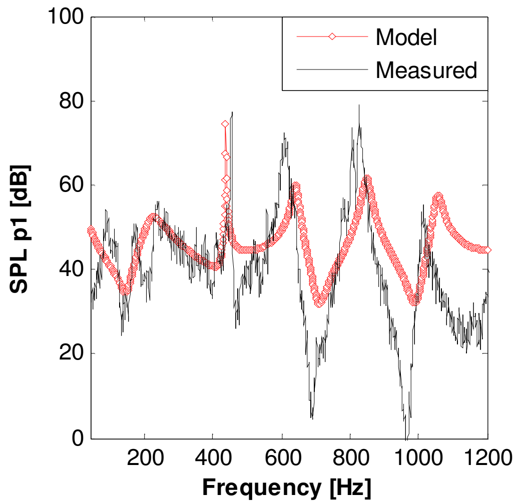

Figure 6: Comparison of measured and simulated pressure spectra at location of P1 for 15.7_1 case

The Result for 15.7_2 case is shown in Figure 7. Despite the operating points are the same, the effect of the other parameters i.e. temperature and ratio between combustion air and cooling air to the main instability frequency can be seen clearly. The first measured instability in Figure 6 is located at 450 Hz, whereas for operating point 15.7_2 it is shifted to a frequency of 410 Hz, see Figure 7. For this case the acoustic network model predicted the instabilities at 420 Hz. High frequency instabilities are also predicted well with the network model.

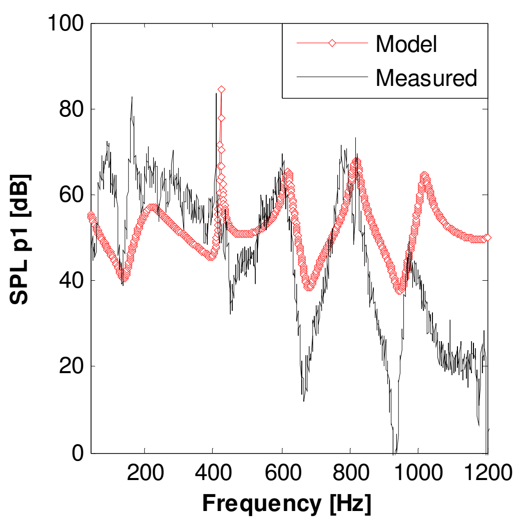

Figure 7: Comparison of measured and simulated pressure spectra at location of P1 for 15.7_2 case

Figure 8 shows the result for case 30.5. For this case the operational power and pressure are twice as high as in case 15.7 where the main instabilities are recorded with a clear peak around 450 and 410 Hz respectively. For the 30.5 case a few peaks are measured between 433 and 450 Hz. The acoustic network model predicted the main frequency of instabilities at exactly 433 Hz.

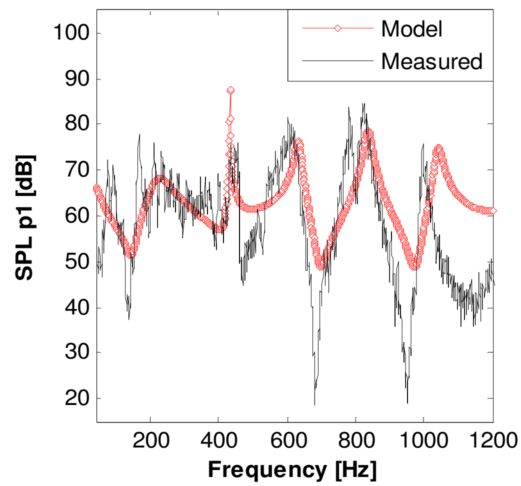

Figure 8: Comparison of measured and simulated pressure spectra at location of P1 for 30.5 case

The acoustic first eigenfrequency of the test rig was calculated around 220 Hz. This frequency was not visible in both the experiment and the simulation. Experiments showed that the dynamics of the flame is around 410-455 Hz. Consequently, the flame dynamics and the first eigenfrequency of the test rig need to coincide to achieve a limit cycle condition. Thus, geometrical tuning in combustion chamber is needed to effect this situation. For this purpose a different combustor dimension is used to model a new liner. Figure 9 shows the result for a shortened liner configuration (0.9 m) in comparison with the original long liner (1.810 m).

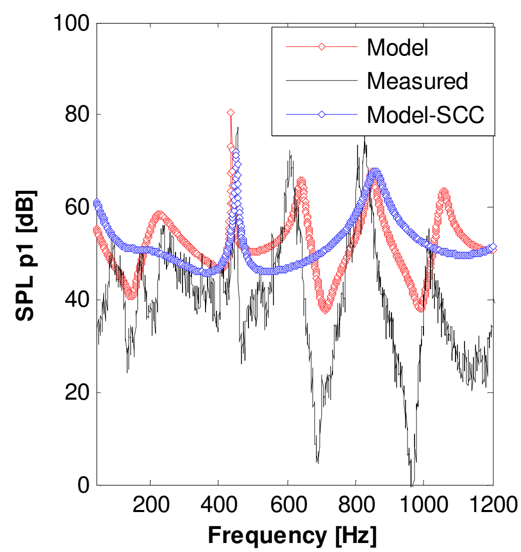

Figure 9: Simulation of different length of combustion chamber (SCC is short combustion chamber)

Conclusions

Measured and modeled are the acoustic response of the combustion system to fuel mass flow fluctuations. The experiments are done for different power settings (125 kW and 250 kW) and at elevated pressure values (1.5 bar and 3 bar). Measured and modeled FTFs are implemented in the network model and comparisons are made. As measurements, due to a MOOG valve restriction, are limited to a frequency of 400 Hz, the acoustic network model is applied to predict the system behaviour at higher frequencies. Finally, the measured and predicted dynamic behaviour in the combustor for different operating points are compared. A very good agreement is found between measured and simulated pressure fluctuations. The results indicated the network modeling approach is a promising design tool for gas turbine combustion applications.

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