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INFLUENCE OF DAMPERS ON THE DAMPING RATES OF A LAB SCALE ANNULAR COMBUSTOR

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The impact of dampers on the stability margin of an atmospheric lab-scale annular combustor is investigated experimentally. In this study, the number of dampers as well as their circumferential distribution is varied. The used acoustic damping devices are quarter wave tube resonators. Furthermore, a modeling approach based on linearized Euler equations is applied to analyze the effect of different resonator arrangements on the stability of the annular combustor numerically.

Damping rates are extracted from dynamic pressure measurements in the combustion chamber that are used to evaluate the damping performance of different resonator arrangements. Two methods are employed for the data analysis. The first one uses Lorentzian fitting of the pressure spectra of the first azimuthal mode. The second method analyzes the autocorrelation of the acoustic pressure signals. Both methods rely on turbulent combustion noise measurements. Furthermore, the azimuthal pressure field is decomposed into two independent modes. This allows analyzing the unbalanced damping effect of spatially asymmetrical resonator distributions on both modes. We see that the circumferential placement of the dampers has a significant influence on the damping of the first azimuthal mode as both decomposed modes need to be damped sufficiently in order to obtain a more stable system. For this reason, it can be shown that with the same number of dampers different damping rates can be observed.

The experimental findings are confirmed by a validated model of the annular combustor that is applied to numerically investigate and assess the influence of the considered resonator configurations on the modal dynamics of the first azimuthal mode.

Keywords: thermoacoustic instabilities, annular combustor, dampers

1. Introduction

Passive damping devices are commonly used in modern day gas turbines to suppress combustion instabilities. These instabilities can otherwise lead to high pressure fluctuations in the system. One consequence of those fluctuations is increased component wear to the point of mechanical failure of certain elements, especially liners. Another consequence of these pressure fluctuations are increased emissions which are one of the key development drivers nowadays.

To prevent instabilities and to estimate the need of dampers, design tools are used. However a more systematic understanding is needed on how to control dampers and on how to apply them in such a way that they are most effective.

Concerning placement strategies for dampers, Bothien et al. [1] (at a full-scale test engine) and Zahn et al. [2] (at a cold flow annular test rig) investigated the placement of dampers (multi volume Helmholtz resonators and quarter wave tube respectively). Stow and Dowling [3] examined the optimal placement of Helmholtz resonators depending on the mode order to attenuate. Azimuthal staging of fuel can also stabilize or destabilize systems, this was investigated by Noiray et al. [4]. Moeck et al. [5] analyzed azimuthal staged heat release sources (i.e. approximated burners) in an annular Rijke tube. These studies give basic principles for properly placing resonators to damp instabilities.

The topic of this paper is how different numbers of quarter wave tubes affect the damping properties of a lab scale annular combustor. The following subjects are covered: The current configuration of the test rig is presented briefly. After that, the methods to determine the damping rates of stable operation points are covered. Five different spatial damper arrangements are investigated for their effectiveness concerning the ability to increase the stability margin. The results are also compared to a numerical model which was used to reproduce the configurations from the experiments.

2. Experimental Setup

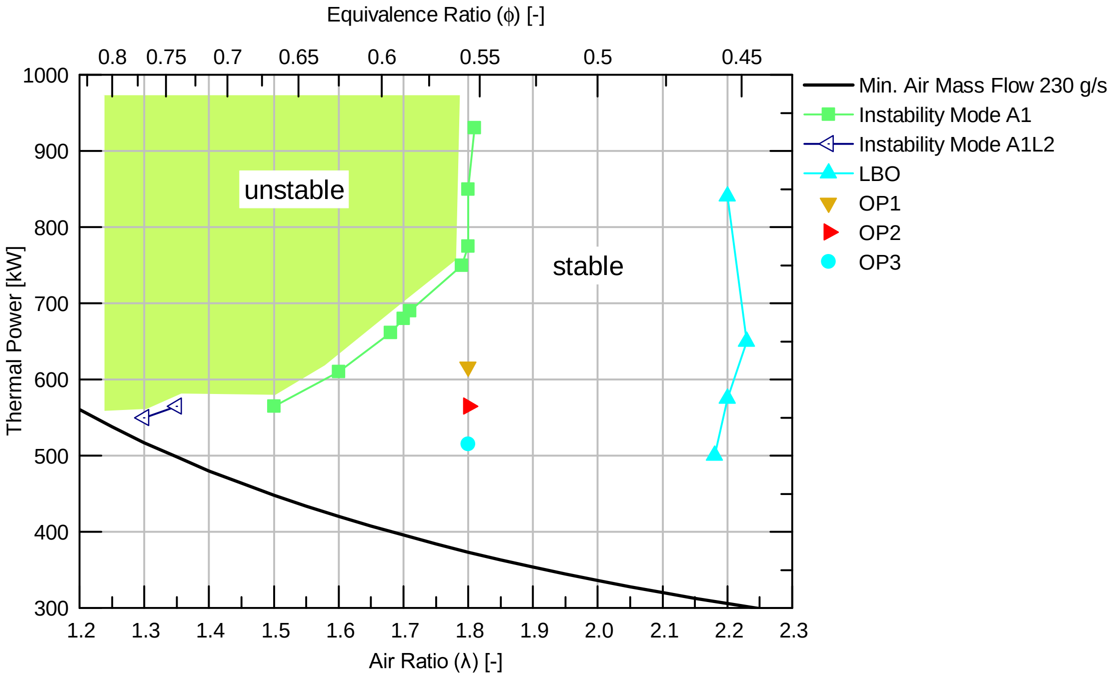

Figure 1: Stability Chart of the Test Rig.

The experimental investigations presented in this paper are conducted at an atmospheric annular combustor test rig. Electrically preheated air enters the plenum via twelve inlets. In perfectly premixed cases – solely such are presented here – the fuel (natural gas) is injected upstream of the plenum. The mixture enters the combustion chamber via twelve equally distributed burners with axial swirl generators. The test rig – combustion chamber and plenum – is made from high temperature steel and is operated at an elevated wall temperature of around 800 K which leads to more engine-like conditions. This design leads to lower heat losses and does not rely on film cooling inside the combustor. Furthermore twelve converging exit nozzles represent turbine inlet-like conditions i.e. the flow is accelerated to high Mach numbers. Thus the test rig is operated in the range of 0.2 - 0.5 bar overpressure. The acoustic pressure is measured using dynamic pressure transducers which are distributed axially and circumferentially around the combustor. One pressure transducer measures the dynamic pressure inside the resonator cavity.

Figure 1 shows the stability map of the test rig without dampers. The test rig exhibits a self-excited instability at higher thermal power and richer mixture. The investigations presented in this paper were done using stable operation points (OP) always investigating the first azimuthal mode. OP1 is defined by a thermal power of $P_{thermal}$=617 kW / λ=1.80. As it is the closest OP to the stability border, lower damping rates are expected. OP2 is $P_{thermal}$=565 kW / λ=1.80. Experiments at lower and higher system pressure are presented. The higher pressure was induced in order to reach the same burner exit velocity as in OP3. OP3 lies at $P_{thermal}$=515 kW / λ=1.80 and should lead to the highest damping rates.

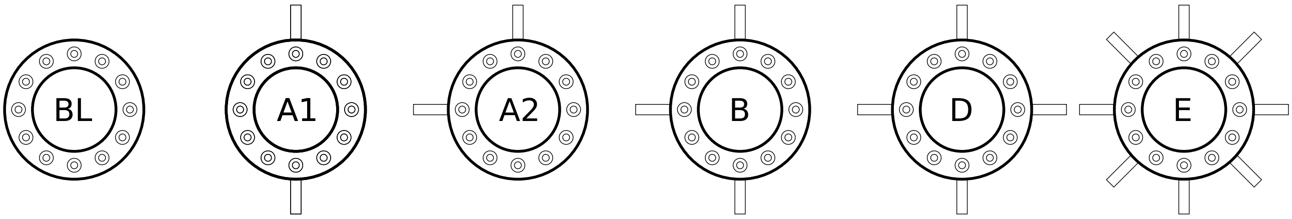

Figure 2: Quarter Wave Tube Configurations Investigated in This Study.

The dampers which are used in this investigation can be mounted to the outer shell of the annulus: two axial positions with 24 ports each are available, at the burners and in between the burners, see Fig. 2 for a sketch. The dampers that are used in this study are modified quarter wave tubes (QWTs): The end facing the combustor is a perforated plate with a porosity of 17.64 %, the tube diameter is 25 mm. Preheated purge air is introduced in the middle section of the cavity to regulate the temperature to plenum-like temperatures as it is the case in real gas turbines. A purge air mass flow of 15 l_n/min/damper is used to prevent hot gas ingestion at the perforated plate which would lead to a detuning of the resonators. The cold characteristics of the resonator are not changed by this amount of purge air. The dampers are designed to reach maximum efficiency at 570 Hz, which is the frequency of the first azimuthal mode in this test rig.

A more detailed description of the test rig and the resonators can be found in Betz et al. [6].

3. Measuring Methods

As already mentioned damping rates are determined in the stable region of operation. For this purpose two methods are used in this study. Both rely on turbulent noise excitation and are briefly summarized in the following:

3.1 Lorentzian Fitting

In the frequency range of interest, a Lorentzian curve – cf. Eq. (1) – can be fitted to the Power Spectral Density (PSD) [7]. From the fitted parameters, the eigenfrequency of the mode (ω_{Res}) and the damping rate (half width at half maximum, α) can be obtained [8]. β/α is the maximum value of the fitted curve at the eigenfrequency. As the system approaches the stability limit, the peak in the power spectrum becomes more distinct and the half width at half maximum decreases.

$$L(\omega) = \frac{\beta\alpha}{\alpha^2 + (\omega - \omega_{Res})^2} \tag{1}$$

3.2 Autocorrelation Fitting

Stadlmair et al. [9] further developed a method originally proposed by Lieuwen [10] for determining damping rates analyzing the autocorrelation of a time series of dynamic pressur data. The Wiener-Khinchin Theorem [11] defines the relation between the PSD of a signal and its autocorrelation function. Equation (2) shows the Autocorrelation function whose parameters (damping rate α, frequency ω and a time step τ) are fitted for each mode η_n based on a Bayesian network approach using a Gibbs sampler.

$$k_{\eta_n\eta_n}(\tau) = \exp(-\alpha_n\tau)\cos(\omega_n\tau)\,. \tag{2}$$

This enhanced method was validated in [9] and applied in [6]. In this study, the results from this method will be compared to the results obtained via Lorentzian fitting once again.

3.3 Mode Decomposition

Instead of the dynamic pressure signals of each sensor, only two decomposed modes (η_1 & η_2) are analyzed. This is possible due to the high number of dynamic pressure sensors located at the outer combustor shell. The dynamic pressure field in the circumferential dimension of the annulus is approximated by Eq. (3) with Eq. (4) and (5) [4, 12]. The two modes η_1 & η_2 are 90° phase-shifted

$$p'(\theta, t) \simeq \eta_1(t)cos(n\theta) + \eta_2(t)sin(n\theta) \tag{3}$$

with

$$\eta_1(t) = A(t)cos(\omega_n t + \varphi_1(t)) \tag{4}$$

and

$$\eta_2(t) = B(t)cos(\omega_n t + \varphi_2(t)) \tag{5}$$

A, B, φ_1 and φ_2 are assumed as slowly varying compared to the angular frequency ω_n, which is a valid assumption for the dynamics in this test rig. n is the order of the mode.

4. Experimental Results

Fig. 2 shows the damper arrangements which we investigated in this study. Different numbers of dampers (2, 3, 4 and 8) were tested. To some extend, the impact of the damper location was also investigated: Two quarter wave tubes were either placed 90° apart or 180° apart. The configuration with 3 dampers was also not evenly spaced: the resonators were placed 90° apart.

Fig 3 shows the damping rates obtained at 4 operation points compared to the baseline case without dampers. It can be observed, that with each damper configuration an increase in damping rate is achieved. One configuration seems to be an outlier: Configuration B with 3 QWTs at OP1 shows damping rates that are even below the baseline. This trend cannot be observed at other operation points. Therefore this measurement is not considered in the analysis. A probable explanation for this is, that during this measurement the real operating conditions differed from the desired OP, so that the real OP was closer to the stability border.

With increasing numbers of dampers a general trend to higher damping rates can be observed.

It can be seen that with the configuration of 2 QWTs 180° apart (A1) and the 3 QWT (B) configuration the decomposed modes η_1 and η_2 are not equally damped. This mode split is due to the fact, that the pressure antinodes of the modes are 90° shifted and thus can't be addressed equally by the dampers. This effect can best be seen if configuration A1 is compared to the baseline case (BL): One mode (η_1) is damped comparably to both modes of configuration A2 (in some cases there is even stronger damping) while the other mode (η_2) remains at the level of the baseline case. In the A1 configuration, two dampers attenuate one mode, while the other one isn't affected at all. Compared to this, in the A2 configuration, there is one damper for each mode and the mode split is non-existing.

The same effect can be seen with the B configuration: one mode is attenuated by two dampers while the other mode is only affected by one damper. The result are diverging damping rates of the modes η_1 and η_2 which tend to configuration A2 and D respectively: The higher damped mode is in the range of configuration D (because it is attenuated by two dampers) while the weaker damped mode is in the range of configuration A2 (because it is only attenuated by one damper).

This can also be seen in the numerical eigenvalue analysis which is presented in section 5.

Considering the quality of the fitting methods, configuration BL and E shall be compared: configuration E is characterized by a high impact of the dampers on the power spectrum of the signal.

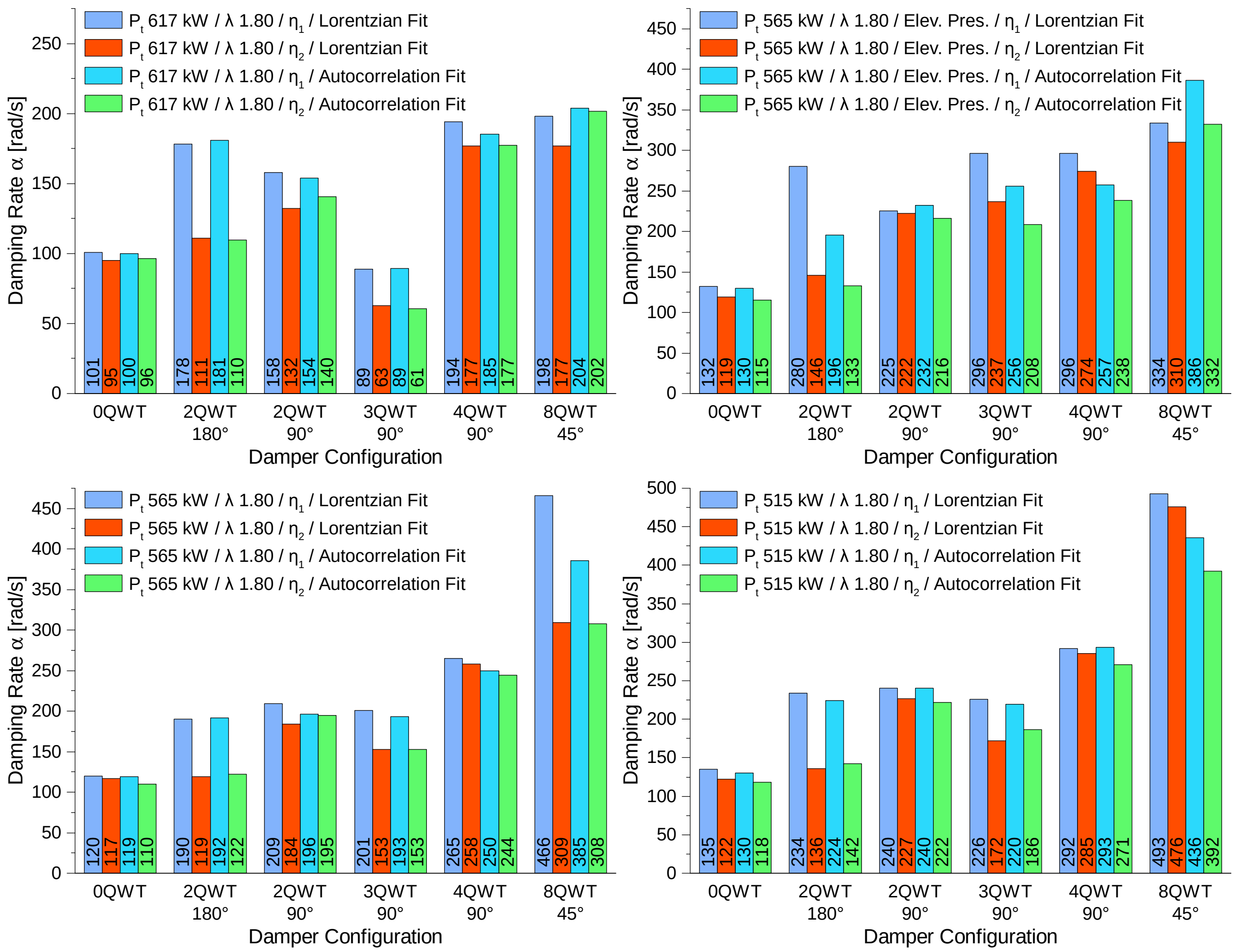

Figure 3: Damping Rates of the Modes η_1 and η_2 Showing the Impact of Different Damper Arrangements at Four Operation Points.

This was already pointed out earlier [6], however no comparison of the methods was done there with cases of high damper impact. Comparing the baseline case results, it can be seen that both methods produce very similar damping rates as the spectrum follows a Lorentzian curve very closely. Concerning the results of configuration E, a higher discrepancy between the results of the two methods can be observed. As it was shown in [6], fitting the Lorentzian function to the PSD has weaknesses in the case of high damper impact. The autocorrelation method is of a more robust nature, partially due to a higher number of supporting points and due to the robust fit of the Gibbs sampler. At such high damping rates the standard deviation can reach the double-digit range. Nevertheless the results of the autocorrelation fit are deemed more reliable especially in the case of high damper impact.

5. Numerical Analysis and Results

A numerical modeling procedure which is presented by Zahn et al.[13], is applied to predict linear stability of the annular combustor that is equipped with the different considered damper arrangements. The general approach of the modeling methodology requires an individual characterization of all combustor components, see Fig. 4. In terms of the investigated annular combustor, the swirl burner characteristics (BTM), the acoustic properties of the damping devices, and the acoustic behavior of the combustor exit nozzles as well as the swirl flame dynamics (FTM) have to determined experimentally and by further numerical approaches that are discussed in more detail in [13].

In scope of the numerical analysis only the three-dimensional domains of the plenum and the

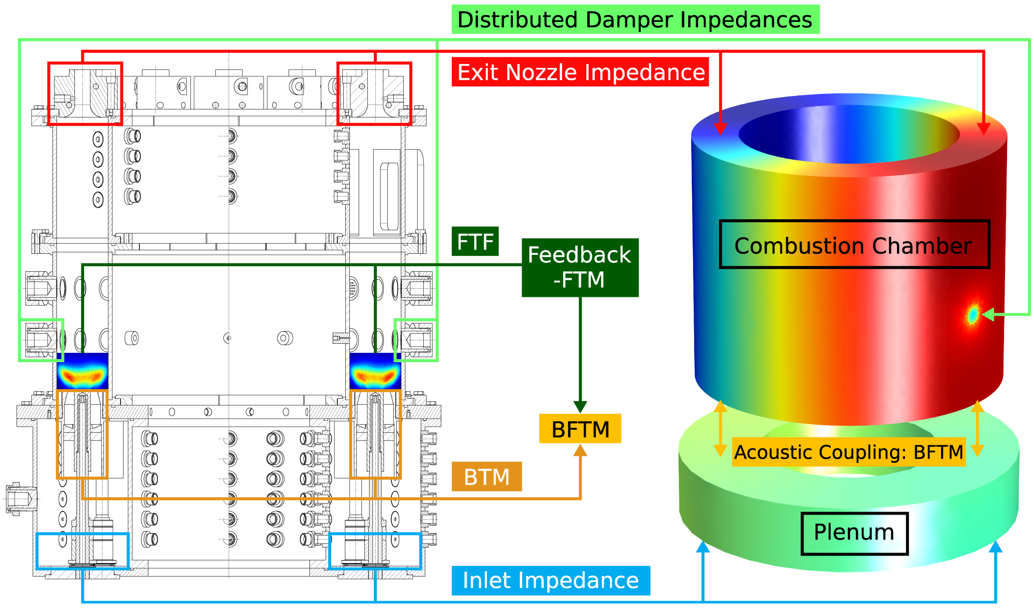

Figure 4: Sketch of applied combustor model and computational domain [13].

combustion chamber are spatially resolved that are acoustically coupled, as shown in Fig. 4 (right). All priorly characterized combustor components are incorporated into the combustor model by impedance boundary conditions and scattering matrices (cf. BFTM). This yields an efficient procedure, although all combustor components are represented accurately. To evaluate linear stability of the annular combustor equipped with the different considered damper configurations as illustrated above, eigenfrequency analyses are conducted on the basis of the linearized Euler equations (LEE) with respect to the considered acoustically coupled computational domains. A stabilized FEM approach is used to solve the LEE in frequency space (see e.g. [2], [13]). The computations are performed on an unstructured grid consisting of approx. 108000 linear tetrahedral elements (i.e. 85000 DOF). From the eigenfrequency analysis one pair of modes is obtained for the first azimuthal mode, denoted as mode 1 and mode 2. The characteristic complex eigenfrequencies of both modes describe the modal dynamics of the first azimuthal mode in terms of "physical" (real-valued) eigenfrequencies and damping rates, see [2]. The mode shape of one mode of the pair with respect to the first azimuthal mode in the combustion chamber is shown in Fig. 4 (right).

In Fig. 5 the numerically determined damping rates for Mode 1 & 2 are compared to the experimentally determined damping rates η_1 & η_2 for OP1 & OP3. The trends derived from the experimental data can also be observed in the numerical results:

- The model shows a nearly linear trend in damping rate increase for optimal damper distributions (A2, D, E) as a function of the damper number compared to the baseline case.
- The mode split which manifested in the experiments (configuration A1 & B) is also present in the results of the model.
- The overall trend of the modeling results for the different damper configurations are in good agreement with the experimental results.
- High deviations from experiment to model can be seen with configuration E, the measured damping rates at OP1 are considerably lower than expected.

6. Conclusion and Outlook

The impact of quarter wave resonators and their spatial arrangements on the damping rates of a lab scale annular combustor was presented. The purposes of this study are briefly summarized before

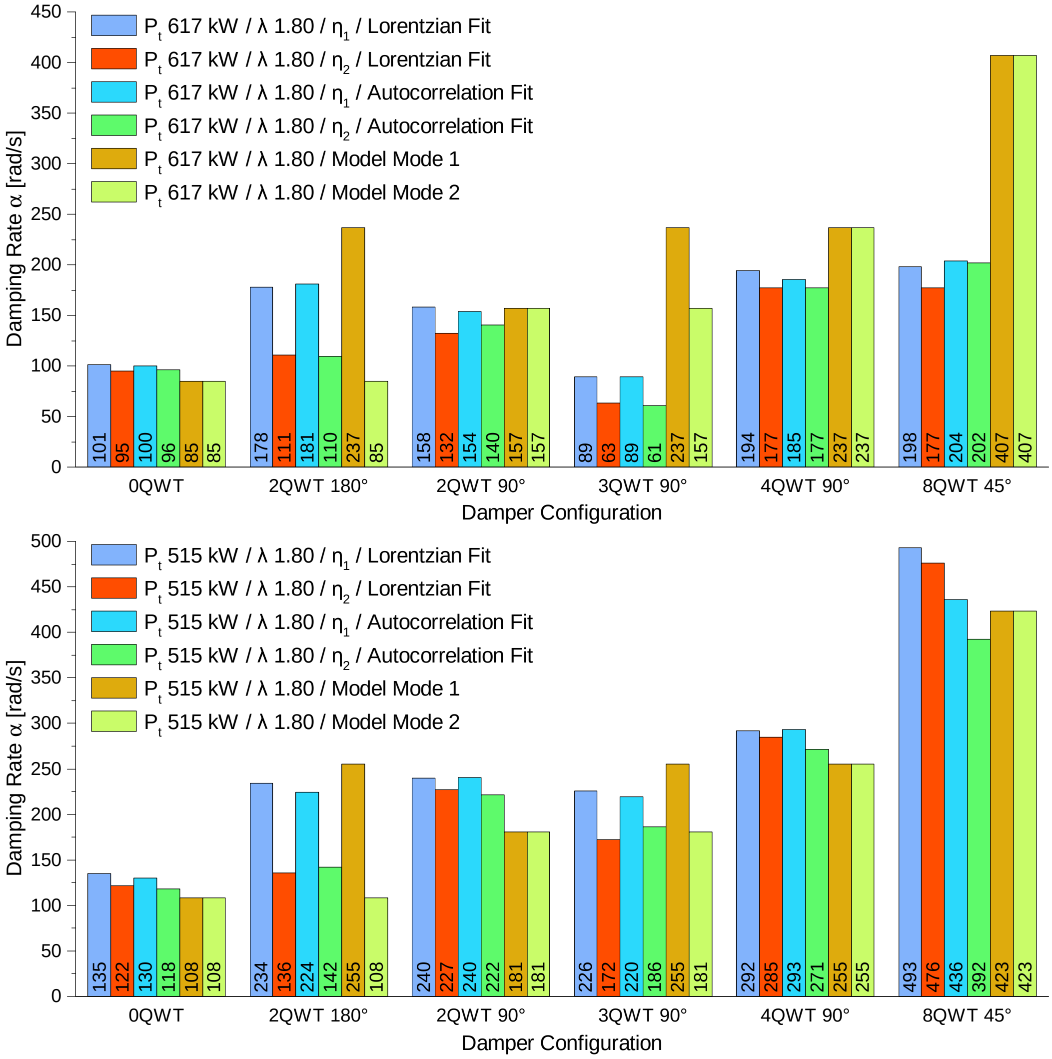

Figure 5: Damping Rates (Experimentally and Numerically Determined) of η_1 & η_2 and of the Modes 1 & 2 Respectively.

the conclusions are drawn:

- Damping rates for varying numbers of quarter wave tubes were measured quantitatively.
- Furthermore nonuniform distributions of dampers were assessed and compared to uniform distributions.
- The autocorrelation fit method was validated against the well-known Lorentzian fit method for all data points.
- The data from the examined configurations were used to validate the LEE based numerical model.

From above mentioned observations the following lessons can be learned.

- After choosing quarter wave tube parameters which proved highly effective in the past, varying increases in the damping rate depending on number of dampers and damper distributions could be observed. This can help in determining the necessary numbers of dampers to stabilize a system.
- In order to attenuate the first azimuthal mode, it can be necessary to use asymmetrical damper distributions while geometrically symmetrical arrangements do not stabilize both decomposed modes.
- Furthermore the autocorrelation fit method delivers comparable damping rates while offering a

higher quality fit and is deemed more reliable at high damper impact.
- In general, the model and the experiments match quite well. Higher deviations can be seen for configuration E. In general the model is sufficient to predict the stability of this combustor.

Future work will incorporate a more sophisticated damper model which incorporates the influence of the combustion chamber's dynamic pressure on the cavity temperature via hot gas ingestion. This effect has a strong influence on the characteristics of the damper and its damping capability. More damper distributions will be evaluated at additional operating points.

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