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# COMPUTATION OF INDIRECT COMBUSTION NOISE BY A CAA–METHOD

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**Abstract**

The available theory for indirect combustion noise, which was mainly developed in the seventies of the last century, is reviewed from the standpoint of current methods in Computational Aeroacoustics. The basic idea therefore – the modes of perturbation which a fluid can show – goes back to the fifties. The coupling of these modes is then investigated with respect to a typical combustion system. An energy cascade for the generation of acoustic energy is found. It is fed by non–acoustic perturbations interacting with the mean flow in the turbine stages. Indirect combustion noise is shown to be an important noise source. Finally, the qualitative theoretical results are compared to a simplified numerical experiment. The numerical model needs to be non–dispersive with respect to small scale entropy perturbations. Furthermore, the correct modeling of the flow ranging from nearly resting incompressible to transonic conditions is required. Additionally, the reflections of the combustion system have to be considered to get a realistic response. The theoretically predicted effects are observed with the simulation.

## 1. INTRODUCTION

Core noise of modern aircraft engines and gas turbines is becoming an increasingly important problem. New technologies are necessary to fulfill upcoming emission control requirements. Many of them are suspected to increase the overall noise level of the engine core. A certain extent of so called core noise can not sufficiently be explained by the pure jet noise or the rotor stator interaction of the turbine stages or the direct noise from the unsteady combustion process alone [1]. Cumpsty and Marble [1] referred the part of the sound field, which is originating back to non–isentropic (temperature) perturbations generated by the flame, but excited by the accelerating flow through the turbine stages as *"indirect combustion noise"*. New technologies, as lean fuel combustion and geared main fans with highly loaded turbine stages both probably lead to a growing importance of the indirect noise generation mechanisms [1]. This is especially expected for entropy perturbations in the wake of the combustion chamber. The flow through the turbines carries perturbations which are initially generated by the combustion process, and can be picked up back by the overall system in case of instabilities. But even without a combustion instability, these initial perturbations finally contribute their energy to the overall sound energy in the system. Not only broadband and low frequency sound emission but also structural vibration fatigue may be caused.

Developing a model for the participating effects a flow perturbation can be understood as a superposition of nonlinear interacting modes, as it was first introduced by Kovásznay [2] and analyzed by Chu and Kovásznay [3] in detail. The definition of these modes is artificial to some extent. Different authors use slightly different definitions, therefore. However, the concept of these modes and the analysis of their interaction helps to gain insight to perturbed flow fields in general. Even though Chu and Kovásznay [3] probably did not consider this, one can read the effect of a large variation of the mean flow from their analysis. Strong flow gradients are found in the combustion system of gas turbines and there would be a first order interaction of all modes of perturbation, then. Usually the combustion chamber together with the turbine forms a complicated system, in which the effects, that contribute to the overall sound radiation are not easy to access. To understand the basic mechanisms which contribute to the noise generation a simplified experiment is set up by DLR Berlin.[2]

The current work first motivates the numerical investigation by the theoretical aspects of the sound energy cascade in a combustion system. Experimental set-up and mathematical model for the numerical investigation are presented shortly in the following sections. Then the numerical method based on the Linearized Euler Equations is applied to compute the sound emission of an entropy perturbation passing a transonic nozzle and the results are discussed in comparison to the experiment. Finally conclusions are drawn.

## 2. MOTIVATION

This section theoretically accesses the unsteady heat supply as noise source. The first subsection described the modes of perturbation used in our analysis. The second concerns the direct generation of acoustical perturbations by a unsteady heat supply. The third subsection refers to the indirect sound produced by accelerating hydrodynamic flow inhomogeneities. The section closes with a discussion of the energy cascade found in the turbine stage at typical operation conditions with choked flow.

### 2.1. The Modes of Perturbation in a Fluid

Based on the concept of Chu and Kovásznay [3] all perturbations are split into acoustic, vortical and entropy mode waves first. Different from their work, we use a resting frame of reference, the vorticity perturbation is assumed to be isentropic and the dissipation effects are not considered to be part of the undisturbed modes. The current state of a instationary moving fluid can always be seen as a perturbation about a mean state.

$$\Phi = \Phi_0 + \Phi' \tag{1}$$

Where $\Phi = \{\varrho, \vec{u}, p\}$ is the state of the fluid. The zero denotes the mean state and the primed quantity denotes the perturbation respectively. The definition of the mean state is not fixed at

[2]The authors gratefully acknowledge the important work of Friedrich Bake, Ulf Michel and Ingo Röhle, who contributed the experimental background for the current work.

this point. A definition of the mean flow state as a stationary solution of the nonlinear Euler Equations will be used later.

The whole state of the fluid is described by the Navier-Stokes Equations. With the perturbation approach (Eq. 1), it is possible to derive a coupled system of equations:

$$\frac{D\vec{\Omega}'}{D\,t} = q_\Omega \tag{2}$$

$$\frac{D\,s'}{D\,t} = q_s \tag{3}$$

$$\frac{1}{c^2}\frac{D^2 p'}{D\,t^2} - \nabla^2 p' = q_p \tag{4}$$

The dissipative terms are considered as sources herein and the material derivatives $D/D\,t$ are taken in a resting frame of reference relative to the fluid. $\vec{\Omega}$ denotes the rotation of the velocity vector and $s$ denotes the entropy per fluid volume. Mathematically these three equations are inhomogeneous wave equations. It describes four modes, which are in the linear sense decoupled in case of a constant mean flow as it was assumed by Chu and Kovásznay [3]. The three modes couple in the linear sense for a non–constant flow, as it was shown for instance by Ffowcs Williams and Howe [4] for low Mach number flows or Marble and Candel [5] for the potential flow in a compact nozzle. The source terms contain the first order coupling of the three modes by the mean flow gradient, as well as dissipation and the nonlinear interaction of modes. The three equations physically describe the transport of vorticity with the flow (Eq. 2), the transport of entropy with the flow (Eq. 3) and downstream and upstream propagation of sound waves at the speed of sound relative to the fluid (Eq. 4).

### 2.2. Direct Sound

We now consider a simplified model for the combustion process, which only adds heat to the fluid, neglecting any mass and force like sources. Unsteady heat supply is the most significant unsteady source with contribution to combustion noise. Other sources like the vortex shading by the inflow of the propellant and air into the combustion chamber through a nozzle or the heat conduction or transmission are neglected in the following analysis, therefore. All quantities assumed to be non–dimensional by the far field density and speed of sound. The direct sound generation due to the heat supply is proportional to the second derivative of the unsteady heat supply in time. This is only partially true, when considering technical combustion systems. The pressure in a fully enclosed constant fluid volume raises with the temperature. This is what Chu and Kovásznay [3] refer to with the statement *"a boundary condition can change"* everything. However, most of the current gas turbines would be more like an open duct than an enclosed volume. Therefore, we do not consider these enclosure effects. The heat supply adds energy to the acoustic as well as the entropy mode. An unsteady spatially constant heat supply adds [3]:

$$q_\Omega = 0; \quad q_s = \frac{\dot{q}}{\varrho_0 C_p T_0}; \quad q_p = \frac{D}{D\,t}\frac{\dot{q}}{\varrho_0 C_p T_0} \tag{5}$$

The source terms represent both effects mentioned above. As expected the entropy perturbation, which is generated by an unsteady combustion is proportional to the change of the rate of heat input.

### 2.3. Indirect Sound by Accelerated Flow Inhomogeneities

The so called indirect sound emission of a combustion system is generated by accelerated entropy perturbations. The source terms for this effect can be obtained from [3] taking the acoustic mode to be a first order potential flow. Therefore the linear interaction of the modes with a potential mean flow are approximated by:

$$q_\Omega = \vec{\nabla} s' \times \vec{\nabla} p_0; \quad q_s = -\vec{v}_0 \cdot \vec{\nabla} s'; \quad q_p = \frac{D}{D\,t}(s'\,\vec{v}_0) \tag{6}$$

The acceleration of a flow inhomogeneity in general generates sound waves, as the non-isentropic flow perturbation assumed above does. Ffowcs Williams and Howe [4] even refer to the effect of an entropy perturbation generating sound, when accelerated in an inhomogeneous flow field as *"bremsstrahlung"*. The first source term $q_\Omega$ is adapted from Crocco's law and describes the generation of vorticity by the interaction of entropy perturbation and base flow. In the following section we will discuss the noise generation by this vortical perturbation.

### 2.4. Energy Cascade

In a realistic turbine cascade the remaining entropy perturbation together with the vortical perturbation, which has been generated in the stage before, pass the next stage. The well known [6] source term of the vortical perturbation can be obtained from [3] to:

$$q_\Omega = -\frac{\partial \Omega_i\, v'_{0,j}}{\partial x_j} + \Omega'_j\,\frac{\partial v_{0,i}}{\partial x_j}; \quad q_s = 0; \quad q_p = 2\frac{\partial^2 (v_{0,i}\, v'_j)}{\partial x_i\, \partial x_j} \tag{7}$$

Both hydrodynamic modes of perturbation add part of their energy to the sound field. That finally leads to an energy cascade, that adds energy from the initial perturbations to the sound field in every turbine stage. As the phase is important for the resulting attenuation or amplification of acoustic energy, there is a chance to reduce noise, as well. The only other way to loose energy from the hydrodynamic perturbations are dispersive and dissipative effects.

## 3. EXPERIMENTAL SET-UP TO INVESTIGATE INDIRECT COMBUSTION-NOISE

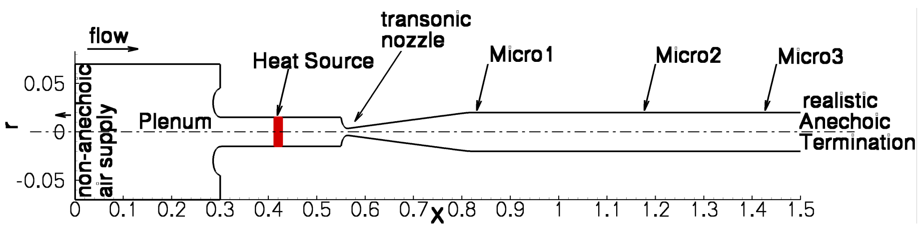


Figure 1. A sketch of the Entropy Wave Generator (EWG) set-up.

The experimental set up is the Entropy Wave Generator (EWG) by DLR Berlin, Bake *et al.* [7] sketched in Fig. 1. It is designed to investigate the source mechanism of indirect combustion noise. To have reliable source conditions, the heat source is realized by electrical heating over thin platinum hot wires. This guarantees a very low time constant of the hot wires and allows the heat source signal to be varied nearly arbitrarily up to the range up to 1 kHz, therefore. Other parameters like the distance between entropy source and nozzle and the mean flow speed can be varied, as well. The experimental set up is described by Bake *et al.* [7]. For the current

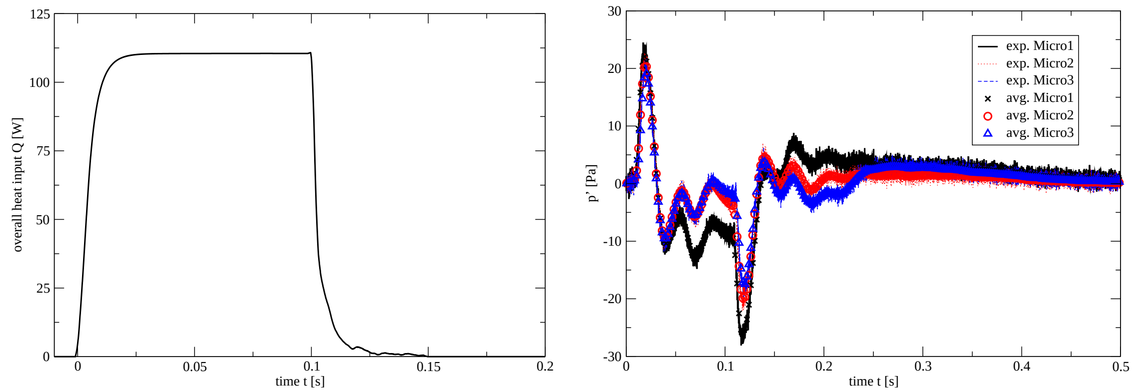


Figure 2. Heating power obtained through the wire temperature measured by a phantom source and resulting pressure at three microphone positions in the exhaust pipe (Experiment by DLR Berlin).

numerical investigation we consider only the unsteady heat source definition shown in Fig. 1, with fixed flow and geometry parameters. The measured heat emission of all hot wires over the time (left) together with the resulting measured microphone signal at three positions in the exhaust duct (right) are shown in Fig. 2. For the numerical investigation the heat is assumed to be spread spatially equalized over the whole volume containing hot wire modules (shown red in Fig. 1).

The flow through the circular pipe is generated by a compressor. The mean flow profile through the Entropy Wave Generator (EWG) shown in Fig. 3, is obtained as a Perturbed Non-linear Non-conservative Euler (PENNE) solution, using the numerical method described below. It is used as unperturbed state in the numerical simulations later. The flow Mach number is around $Ma = 0.03$ at the wire position and reaches $Ma = 1.1$ in the section behind the nozzle throat. The flow is choked by a shock a few millimeters behind the nozzle throat. Apart from the axisymmetric design and the relatively high Mach number at the heat source the configuration is comparable to a technical combustion chamber.

## 4. MATHEMATICAL MODEL AND NUMERICAL METHOD

The numerical method used here is adapted from a method to describe the sound propagation in aero-engine intakes [8]. The governing equations can be chosen ranging from a system representing the wave equation over the non-isentropic Linearized Euler Equations (LEE) to a Perturbed Nonlinear Non-conservative Euler Equation (PENNE) as it was described by Long [9]. In the current investigation we use an axisymmetric model, as the whole system and the source are axisymmetric.

The optimized spatial discretization is based on finite differences of 4$^{\text{th}}$ order and uses the DRP scheme of Tam and Webb [10]. The explicit time stepping of 4$^{\text{th}}$ order uses the alternating 5/6 stage LDDRK of Hu *et al.* [11]. The Extended Helmholtz Resonator model [12, 13] is used to implement the measured termination impedance of the non-reflective termination. The

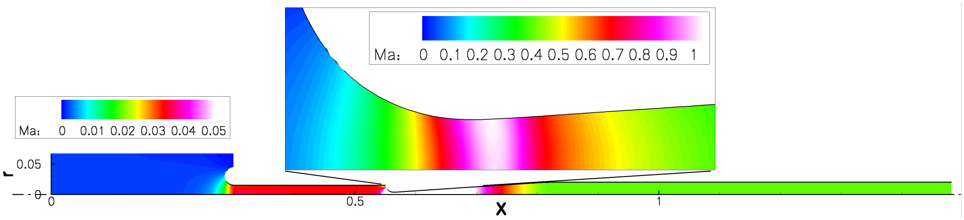


Figure 3. Mach–number of the mean flow through the Entropy Wave Generator (EWG) obtained by a nonlinear Euler simulation.

method is rounded up by a so called Perfectly Matched Layer (PML) [14] boundary condition to implement a non–reflective termination of the computational domain.

To avoid grid oscillations different explicit filter stencils are available, which have different characteristics. The only explicit filter which does not add grid oscillations, when applied to a step function, as the shock is, is a three point explicit filter. Therefore, an adaptive selection of the filtering stencil based on the change of the overall Mach number within the stencil is performed. If the change is larger then a certain predefined threshold the filter stencil size is reduced until the three point filter is reached.

## 5. RESULTS

The numerical mesh consists of 13 grid blocks for parallel computation and to implement the different boundary conditions with overall 72449 grid points. The mesh design is body fitted and as Cartesian as possible, which is also achieved by the structured blocking, The mesh is refined in the nozzle keeping the aspect ratio close to 1. The computation is run over $1 \times 10^6$ time steps to reach 0.4 s in real time. This takes 126 hours on a single P4, 3.0 GHz. Fig. 4 shows the

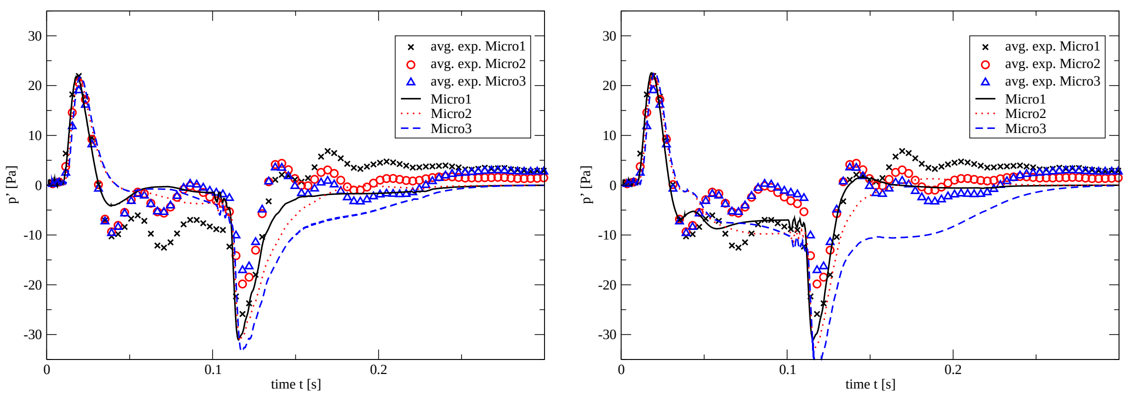


Figure 4. Resulting pressure response at a microphone in the outlet duct without (left) and with measured termination impedance (right).

results of the simulation. Different models for the EWG were tested and are compared directly with respect to the temporal development of the pressure response in the outlet duct. These are an approximation of the measured impedance function and a fully non-reflective open end. The lower end is considered non–reflective and fully reflective, respectively. The best result with respect to the experiment is observed with an anechoic inlet. As the measured termination impedance is close to anechoic, the results differ only in details, if an anechoic inlet is used. There are differences especially for the negative pressure peak, which is excited by the negative temperature gradient passing the nozzle, and interferes with the reflection from the ends. The exhaust pipe is dominated by the indirect sound, which is excited after the first heated air has reached the nozzle. This takes about $0.01$ s for a convection at $Ma = 0.03$ for $0.1$ m between the closest hot wire and the nozzle. Even though the temperature increase of the "hot spot" is only $11$ K ($\approx 4$ % $T_0$), the observed sound pressure amplitude reaches up to $30$ Pa. The direct sound is of much smaller amplitude and reaches only $2$ Pa. The results clearly indicate the observed pressure curve is dominated by indirect combustion noise. Behind the nozzle vortical structures develop, as can be seen in Fig. 5.

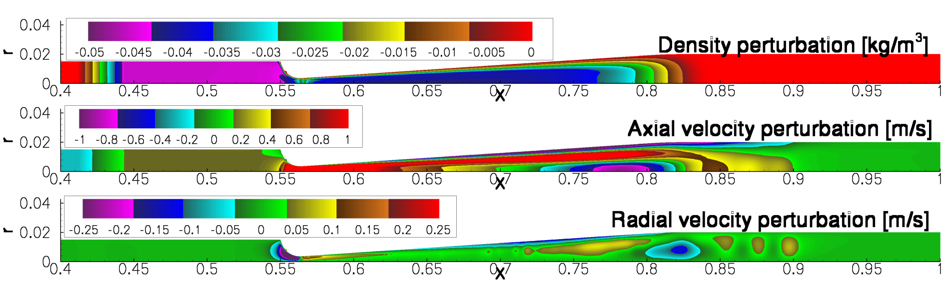


Figure 5. Instantaneous density, axial and radial velocity perturbations show developed vorticies after the nozzle.

## 6. DISCUSSION

The results show, that the presented numerical method is able to predict the response of the combustion system to an entropy perturbation form the flame correctly. The base flow obtained using the PENNE model with the same mesh and numerical method agrees well with the experimentally observed flow conditions. The linear model (LEE) is sufficient to describe the resulting sound field of the electrical heated air, even if the flow reaches transonic conditions. However, it is necessary to use adaptive filter stencil sizes in the region of a shock then, to obtain a stable solution with the finite difference scheme. The generation of vorticity fits qualitatively correctly into the theoretical framework. However, the current method would not correctly predict the vortex transport and dissipation, as the dissipative effects are neglected. The dissipation, turbulent mixing and heat radiation of the entropy perturbation transported between heat source and nozzle has a negligible effect in the EWG. This observation can probably be transferred to many combustion systems using a short combustion chamber. There is an important effect of the numerical boundary conditions. The replacement of a fully non–reflective condition by a measured impedance could improve the agreement between numerical solution and experiment. However, the conditions at the air inflow could not be measured and consequently not be modeled. The measured impedance together with a hard wall here leads to an growing pressure in the system, whereas a combination of fully reflecting plenum inflow and non-reflective outflow would produce large oscillations at the microphone positions. These oscillations have the correct frequency with respect to the experiment and show the correct decay shape, but at a higher amplitude. Therefore, a measurement of the impedance of the lower end seems highly recommended. Alternatively the simulation could include the detailed geometry of the inflow.

## 7. CONCLUSION

The theory for indirect combustion noise is reviewed from the current state of the art in of Computational Aeroacoustics. It is shown theoretically and in a numerical experiment, that indirect combustion noise may be an important noise source. The observed sound pressure levels are experimentally supported. The numerical results show, that the theoretically predicted energy cascade is present even in a simplified model. Additionally it is shown, that the modeled boundary conditions highly influence the observed pressure response to the heat input. Thus, a good model should consider measured impedances.

## ACKNOWLEDGMENT

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