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CALCULATING THE TURBULENT NOISE SOURCE OF PREMIXED SWIRL FLAMES FROM TIME MEAN REACTIVE RANS VARIABLES

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Abstract

In the paper we demonstrate the calculation of the noise source of turbulent premixed swirl flames based on a new model using the time mean RANS variables from a reactive CFD calculation: heat release rate, turbulence kinetic energy and dissipation rate. The model is based on an assumed model spectrum for the reaction progress variable, on a premixed mean turbulent heat release model as well as on previous work on turbulent flame noise formation. It includes no free calibration coefficients. The new model has been validated with encouraging agreement using experimentally measured input data and comparing the resulting spectra of acoustic power with measurements on swirling flames varying power, equivalence ratio and fuel composition. Steady reactive RANS- CFD calculations were performed and validated to which the noise model was applied. Very good agreement between the measured and modelled spectra of acoustic sound pressure was obtained.

INTRODUCTION

Over the last years turbulent combustion noise has become a focus of interest in industrial gas turbines, jet engines and domestic water heaters in order to optimize designs for durability and minimal sound emission. The fundamental connection between the turbulent reactive flow field and the generation of sound is known since the pioneering work of Lighthill [8] in 1952. There its mathematical description emerged as a by- product of the analysis of aerodynamically generated noise which most of the early theoretical and experimental work was focussed on (e.g. [10]). Almost a decade later the first studies on combustion generated noise were published (e.g. [2]). In the early 90s Clavin [3] proposed a connection between the turbulent flow field and the noise emission of turbulent premixed flames based on Strahles work [13], that exhibited a reasonable amplitude decay according to an exponential power law but could not provide a quantitative closure for sound pressure levels. Klein and Kok [6] proposed a formal closure for calculating sound pressure from turbulent diffusion flames based on local mean turbulence, mean reaction rate density and an assumed model spectrum for heat release fluctuation. However, it relied on the specification of an empirical coherence length scale to calibrate the model. Also the space time mapping proved to be non-universal. In contrast to the previous work, the model for turbulent premixed flames applied in this paper is capable providing quantitative spectra of acoustic power based on local mean turbulence and heat release rate density without adjustments. Since its derivation is described in detail in Hirsch et al. [5], here we only summarize the model theory before we show and discuss reactive RANS (Reynolds Averaged Navier Stokes) CFD (Computational Fluid Dynamics) calculation results and the comparison of thus calculated sound spectra with experiments.

THEORY

For the thermo-acoustic source term the time derivative of the local heat release density is needed as shown in e.g. Crighton et al. [4]. From the spectral viewpoint this is equivalent to multiplying the frequency spectrum of heat release density with the angular frequency. Therefore the provision of a model for the turbulent heat release density spectrum allows the calculation of the turbulent combustion noise spectrum, e.g. by a Finite Element solution of the resulting inhomogeneous wave equation. For free space an analytical solution exists, which we use below for a convenient comparison with measurements. In the following we only sketch the model presented in Hirsch et al. [5] to obtain a local model frequency spectrum for the heat release density based on mean turbulence quantities, but do not discuss its assumptions.

For sufficiently large turbulence Reynoldsnumber $Re_t = \frac{u' \cdot l_t}{a_0}$, a_0 being the thermal diffusivity, and requiring that the turbulence velocity u' and turbulence integral length scale l_t are sufficiently larger than the laminar flame speed s_l, $u'/s_l > 1$ and the laminar flame thickness δ_l, $\delta_l/l_t \approx a_0/(s_l \cdot l_t) << 1$, the mean heat release rate density $\tilde{q}$ ($[W/m^3]$) can be expressed as a function of the mean scalar dissipation rate $\tilde{\xi}_{c^2}$ of the variance of the progress variable $\widetilde{c'^2}$ [9] and is modelled formally as the dissipation rate of a passive, diffusive scalar, where $\epsilon \approx u'^3/l_t$ is the dissipation rate of turbulence kinetic energery $k = 1.5u'^2$, both per unit mass.

$$\tilde{q} = \rho_0 \, Y_{F,0} \, H_u \, \tilde{\xi}_{c^2} = \rho_0 \, Y_{F,0} \, H_u \cdot C_D \frac{\epsilon}{k} \widetilde{c'^2} \tag{1}$$

with $C_D = 2.0$ as proposed in [9]. $\rho_0, Y_{F,0}$ and H_u are the reactant density, fuel mass fraction and lower heating value respectively. For simplicity it is assumed that the thermal diffusivity of the fresh mixture a_0 be equal to the kinematic viscosity, i.e. $Pr = 1$. Equation (1) suggests that the spectrum of heat release can be obtained if the spectrum of $\widetilde{c'^2}$ were known, thus:

$$\tilde{E}_q = \rho_0 \, Y_{F,0} \, H_u C_D \frac{\epsilon}{k} \tilde{E}_{c^2} \tag{2}$$

Assuming for the moment that $\widetilde{c'^2}$ behaves as a passive scalar, we may use the model spectra derived in Tennekes and Lumley [14] with their proposed constants $\alpha = 1.5$, $\beta = 0.3$.

After some algebra one obtains:

$$\begin{aligned}\tilde{E}_q &= \tilde{q}\cdot\frac{C_s\cdot C_D}{\alpha}\cdot\alpha\,\frac{\epsilon^{2/3}}{k}\,\kappa^{-5/3}\,\exp\left[-\frac{3}{2}\left(\pi\beta\alpha^{1/2}\,(\kappa l_t)^{-4/3}+\alpha\,(\kappa\eta_{c^2})^{4/3}\right)\right] \\ &= \tilde{q}\cdot\frac{C_s\cdot C_D}{\alpha}\frac{E_{u^2}}{k}\end{aligned} \tag{3}$$

where $C_s = \alpha/C_D$ for the limit of large turbulent Damköhler number $Da_t \to \infty$. E_{u^2} is the turbulence kinetic energy spectral density as e.g. proposed by Tennekes and Lumley [14] of which the integral over all wavenumbers is the turbulence kinetic energy k. η_{c^2} is the Corrsin / Kolmogorov scale, which for the passive scalar becomes $L_C = (a_0^3/\epsilon)^{0.25}$. To include the effect of c not being a passive scalar Hirsch et al. [5] consider two effects:

1. The propagation speed of the flame front characterized by the Gibson scale $L_G = s_l^3/\epsilon$ which will tend to ”iron out” the small scales, i.e. produce a spectral cut off at high wavenumbers. They propose to consider it through an effective Corrsin scale:

$$\eta_{c^2} = \max[c_G L_G, L_C] \quad \text{with} \quad c_G = 3.0 \tag{4}$$

2. The change of flame regime. With decreasing Damköhler number or increasing Karlovitz number a decrease in spectral amplitude is expected due to the effects of finite rate chemistry which reduce the variance of the progress variable $\tilde{c'^2}$ ([1],[9]). Reducing the variance corresponds to scaling the peak amplitude of the power spectrum, since the spectral cut-off is already considered. Hirsch et al. use the average heat release density model proposed by Schmid et al. [11] and derive a scaling function for C_s in equation (3) which is:

$$C_s = \frac{\alpha}{C_D}\frac{\tilde{c'^2}(Da_t)}{\tilde{c'^2}(Da_t\to\infty)} = \frac{\alpha}{C_D}\left(\frac{\frac{s_l}{\sqrt{2/3\,k}}+(1+Da_t^{-2})^{-0.25}}{\frac{s_l}{\sqrt{2/3\,k}}+1}\right)^2 \tag{5}$$

$$Da_t = \frac{0.09\cdot k\cdot s_l^2}{\epsilon\cdot C_c^2 a_0} \quad \text{and} \quad C_c \approx 1.2 \quad \text{for Natural Gas fuel} \tag{6}$$

With this scaling function the necessary asymtotic behaviour of the heat release spectrum is obtained, since $C_s \to 0$ for $Da_t \to 0$, i.e. the homogeneous reactor limit, where no heat release fluctuations exist. Up to here we have presented a model to determine the local wavenumber spectrum of heat release density based on average quantities by inserting equations (5) and (4) into equation (3). To obtain the temporal spectrum $\psi(2\pi f)$ from the spatial spectrum $E(\kappa)$ Hirsch et al. follow Tennekes and Lumley [14] by requiring that the spectral energy content of corresponding scales is conserved, i.e. $\kappa\,E = 2\pi\,f\,\psi$ and postulating that the Lagrangian spectrum is a simple rearrangement of the Eulerian spectrum. For homogeneous turbulence this gives the following mapping between wavenumber and frequency.

$$\kappa = \frac{2\pi}{l_t}\cdot\frac{(2\pi)^{0.5}}{\alpha^{0.75}}\cdot(f\,\tau_c)^{1.5} \tag{7}$$

Here $\tau_c = l_t/u'$ is the characteristic time scale. In premixed turbulent combustion this time scale is given by the passing time of the large scales through the flame brush, i.e. the ratio of the flame brush thickness δ_t and the turbulent burning speed s_t: $\tau_c = C_\tau \delta_t/s_t$. Using the turbulent burning velocity relations also proposed by Schmid, Hirsch et al. obtain:

$$\tau_c = C_\tau \cdot \frac{\delta_t}{s_t} = C_\tau \cdot \frac{l_t}{u'} \cdot \frac{a_0/(s_l \cdot l_t) + (1 + Da_t^{-2})^{0.25}}{s_l/u' + (1 + Da_t^{-2})^{-0.25}} \tag{8}$$

with $C_\tau \approx 0.5$. Given the local frequency spectra of heat release density $\psi_q = \kappa E_q/(2\pi f)$ by combining equations (3), (4), (5), (7) and (8) we can now compute the frequency spectrum of the acoustic power in free space according to e.g. Winkler et al. [16]. For this the correlation spectrum of the partial derivative of the heat release with respect to time needs to be integrated over the flame volume V_f. As shown there this amounts to integrating the contributions of the coherent monopole sources with a correlation volume $V_{coh} = 4\pi/3\, l_{coh}^3$ over the flame volume. The coherence lengthscale is evidently $l_{coh} = \delta_t$, which is confirmed by experimental findings [15].

$$P_{ac} = \frac{2\pi}{4\pi\rho_0 c_0}\left(\frac{\gamma - 1}{c_0^2}\right)^2 \int_{(V_f)} (2\pi f)^2 \left(\frac{\kappa E_q(\kappa)}{2\pi f}\right)^2 \cdot \frac{4\pi}{3}(\delta_t)^3 dV_f \quad \left[\frac{W}{Hz}\right] \tag{9}$$

In the following section we shortly present the experimental setup for this paper, before we turn to the results of two reactive RANS CFD cases and compare with experiment.

EXPERIMENTS

To validate the theory presented above, an experimental parameter study, reported in Hirsch et al. [5] was performed on a turbulent premixed swirl jet flame varying power, equivalence ratio and fuel composition at a fixed swirl number of $S = 0.5$. These experiments provided both the input, i.e. heat release and turbulence data, and the output, i.e. the measured noise spectra, to validate the combustion noise model without adding further modelling uncertainty. The excellent validation of the theory encouraged us to apply the model to a RANS CFD simulation.

For the experiments reported here the same modular and flexible swirl burner (see Figure 1) was used [16]. The outer diameter of the nozzle is $D = 40\ mm$ and the diameter of the centerbody is $D_{lance} = 16\ mm$. The fuel was Natural gas (NG) operating with an equivalence ratio of $\phi = 0.83$. A static mixer upstream guaranteed homogeneously premixed air-fuel mixture. The thermal output was constant at $30\ kW$ and swirl number was set to $S = 0$ and $S = 0.8$. The mean axial velocity at the burner exit was $10.5\ m/s$ and the Reynolds number based on the hydraulic diameter of $d = 24\ mm$ of the annular gap was $Re = 15000$. Acoustical measurements were performed with a calibrated microphone intensity probe at $0.7\ m$ distance from the flame. Chemiluminescense was measured with an intensified camera using an interference filter for $308\ nm$. Velocities and turbulence quantities were measured with a high speed particle imaging velocimetry (PIV) system. For space reasons we kindly need to refer the reader to [5, 15, 16] for the details of the measurement techniques.

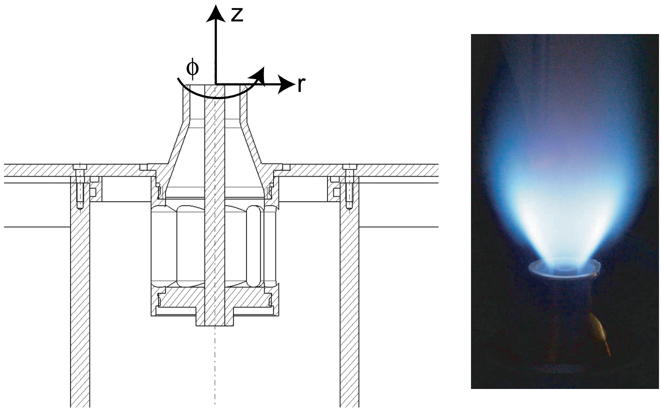

Figure 1: Geometry of the burner burner and typical flame

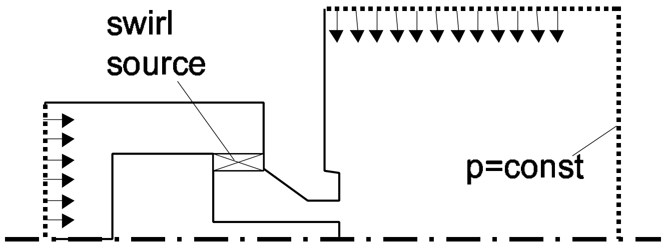

Figure 2: Schematic of the boundary conditions used

CFD MODEL

The swirling turbulent premixed flame with a thermal power of $P_{th} = 30kW$ and $\phi = 0.83$ was calculated for two swirl numbers $S = 0$ and $S = 0.8$ using the CFD code FLUENT V6.1 in the 2d- axisymmetrical formulation using the heat release model by Schmid et al [11] which was modified ad-hoc with a quench factor function based on Damköhler number to suppress reaction in the outer shear layer. This was done to obtain a fair comparison with the experimental flame shape, since the focus of this work was not combustion modelling but the assessment of the noise source model with CFD.

$$\tilde{q} = 4.96\frac{\epsilon}{k}\left(\frac{s_l}{\sqrt{2/3\,k}} + (1 + Da_t^{-2})^{-0.25}\right)^2 \tilde{c}\cdot(1-\tilde{c})\rho_0 Y_{F,0} H_u \cdot \underbrace{(1-\exp[-10Da_t])}_{\text{quench term}} \quad (10)$$

Equation (10) and the noise post-processor, equation (9) were added to the code as User Defined Subroutines and using User Defined Scalars. The calculation domain is shown schematically in figure 2. As shown there it included the plenum upstream of the swirler where the mixture flow rate was specified. The swirler meridional cross section was modelled exactly. The swirl and turbulence created by the tangential swirl ducts were included as source terms in the momentum and turbulence equations as proposed by Kiesewetter et al. [7]. Downstream of the burner exit a region of $25D$ burner diameters axially and $10D$ radially was discretized. On the radial boundary inflow velocities of $v = 0.3\,m/s$ were specified in the case of $S = 0.8$

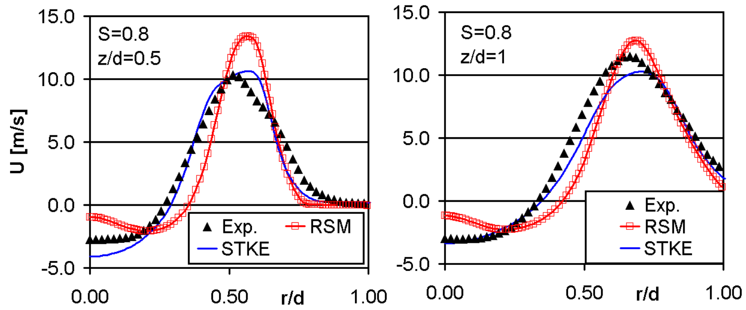

Figure 3: Comparison of axial velocity from experiment and CFD with two turbulence models

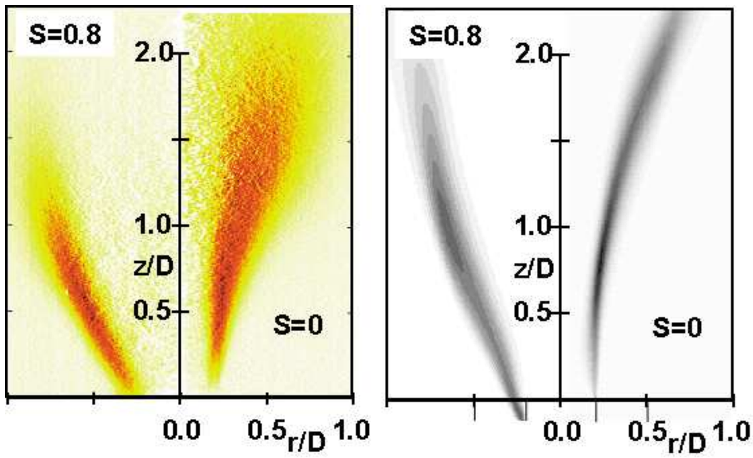

Figure 4: Comparison of flame shapes, left measured deconvoluted chemiluminescense, right mean reaction rate density from CFD

and $v = 0.05\ m/s$ in the case without swirl. Like this the jet entrainment could be satisfied and convergence was improved. A sensitivity study showed that the results in the flame zone did not change with a $\pm 100\%$ of these values. In total 75000 cells were used.

RESULTS

The CFD calculation results for two turbulence models, the Standard $k-\epsilon$ Model (STKE) and the Reynolds- Stress Model (RSM) were compared with the measured velocity data from PIV. In figure 3 the radial profiles of axial velocity are shown at two axial positions downstream of the burner exit for the case of the strongly swirling jet. The comparison is fair for both turbulence models.

In figure 4 we compare exemplarily the experimental shape of the flame shown left, with that of the CFD calculation shown on the right. The experimental data is obtained from deconvoluting the average chemiluminescense pictures while the CFD data results from plotting the heat release rate density. While the calculated flames appear slightly too long, it can be seen that the CFD calculation very nicely reproduces the measured flame geometry. In particular the change of length due to the swirl is captured very well.

Finally, we show the measured and modelled acoustic power spectra in figure 5. The model curves result from applying equation (9) using the CFD data for mean heat release rate

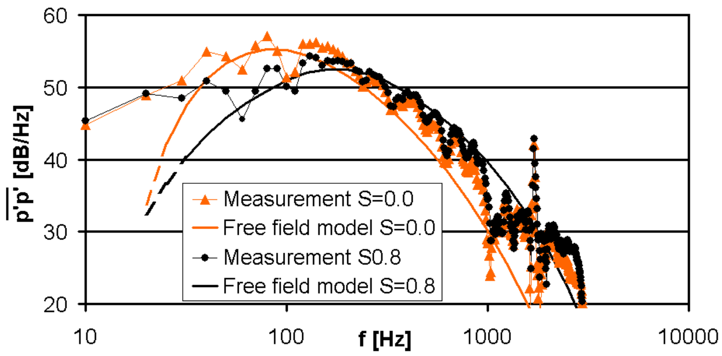

Figure 5: Measured power spectra and calculated spectra from CFD postprocessing

density, the turbulence kinetic energy k and the turbulence dissipation rate ϵ. Considering the obvious imperfection of the acoustical measurement due to room acoustics, the comparison is excellent. The magnitude change and the shift of peak frequency between the non swirling and the swirling flame have been captured as well as the spectral fall off for higher frequencies. The model explains these changes, since the non swirling flame is longer and therefore has larger turbulent length and time scales. This shifts the characteristic timescale in equation (8) to a larger value, i.e. lower peak frequency, and also increases C_s from equation (5), i.e. the peak amplitude, due to higher Damköhler number.

CONCLUSIONS

The results of this paper show that the model proposed by Hirsch et al. [5] can be directly used with CFD models to calculate the thermo acoustic source generated by turbulent premixed flames, if a reasonable mean turbulence and mean reaction rate density is calculated by the CFD model. In particular the proposed mapping between spatial and temporal spectrum as well as the magnitude scale function are essential for the excellent comparison with measured acoustic power spectra. The current results mark the first time that such a calculation resulted in quantitative agreement without the need to adjust empirical parameters of the noise model.

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