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Towards a Novel 0D Gas Turbine Combustor Modeling: Bridging the Gap Between Dimensional and Engine Performance Approaches

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Abstract

This paper aims to present a new 0D combustion model adapted to gas turbines and accounting for the flame dynamics through the awareness of combustion process and turbulence features. Hence, thanks to reduced 3D equations and Large Eddy Simulations (LES) data, this model allows to correctly represent combustion of transient operating conditions, which is unique for a 0D model of combustion in aeroengines. The developed 0D combustion model focuses on the description of turbulent premixed combustion, typically encountered at the injector exit of gas turbines combustor, by using the Coherent Flame Model (CFM) formalism.

1. Introduction

In order to reduce greenhouse gases and pollutant emissions, current aeronautical engine technologies must be improved. Accordingly, new systems like lean premixed burners or breakthrough technologies, such as hybrid architectures or modified and combined thermodynamic cycles, should be deployed. However, continuously improving aeroengine performances while avoiding an increase of design cost requires the enhancement of multi-physics simulation tools. Indeed, even if system simulation is today increasingly used in the design process of an aeroengine, the combustion chamber in those approaches is still represented as a black box where take place thermodynamic and chemical processes. This weak physical awareness involves difficulties in the design process to well predict the engine performances in terms of delivered powered but also in terms of pollutants emissions.

Presently, simulation tools are divided into two main classes having different purposes. On the one hand, dimensional approaches, and in particular 3D simulations, provide detailed features of thermodynamics and turbulence, in consideration for a high simulation time cost. These simulations are commonly used to precisely design a component. On the other hand, 0D/1D simulations provide quasi-real time calculations allowing to test different system configurations and operating conditions. They are used to evaluate the overal performance of an engine and to determine boundary conditions of a system for setting up 3D simulations. However, combustion is simply represented as heat contribution or by 0D chemistry models[1,16–19] . Heat contributions suffer of a lack of chemistry prediction, and 0D chemistry models do not take into account flame dynamics prediction due to flame/turbulence interaction. Moreover, they consider steady-flow assumptions, which leads to a lack of validity in transient operating conditions whereas these conditions considerably affect the overall engine performances.

A solution for enhancing the range of applications of engine performance approaches consists in developing more physical models including a detailed description of the combustion process. However, combustion within gas turbine combustors is a very complex process because of the presence of two kind of regimes : premixed flame in the primary zone near the injector, and non-premixed flame in the intermediate zone downstream the dilution holes of the flame tube, Figure 1. Such models can be obtained either by using phenomenological approaches or by reducing 3D CFD combustion models[14] . This paper presents a 0D combustion model adapted to turbulent premixed flames in gas turbines combustors. This model takes into account the flame/turbulence interaction and the flame dynamics through the reduction of the 3D CFM formalism[2,9] described below. In order to extract consistent observations for

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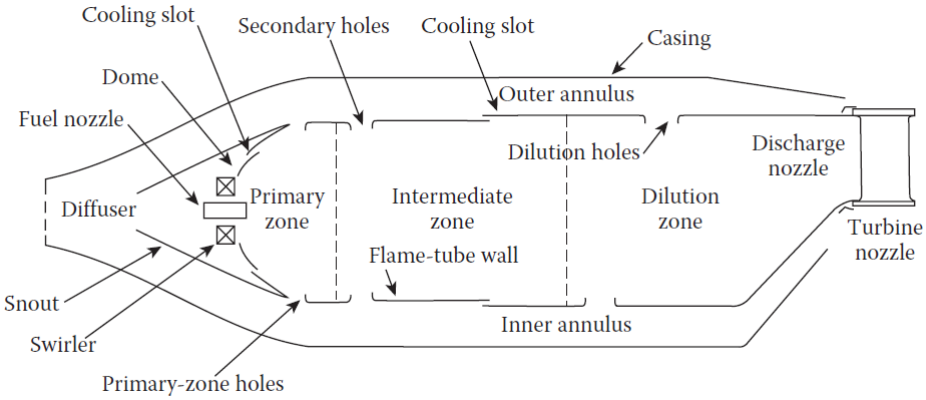

Figure 1: Simplified representation of a conventional combustor[8] .

characterising flame dynamics, 3D simulations are performed on the well-known turbulent premixed combustion academic burner, PRECCINSTA. It can be noticed that the PRECCINSTA burner is chosen because it has been thoroughly analysed by means of experimental studies and numerical simulations. In addition, this burner features a swirl injector similar to real aeroengines injection technologies. In order to be more predictive, the transported equations governing the 0D combustion model are corrected with coefficients which are deduced from LES parametric studies. This original methodology involves the capability to adapt this model to different aeroengine combustor geometries.

The gas mixture is assumed to be made of 12 species, in order to easily introduce the pollutant formation calculation. The 12 species are the fuel with general formulation $C_xH_yO_z$ and N_2, O_2, H_2, H_2O, CO, CO_2, NO, NO_2, Unburnt HydroCarbons (UHC), NH_3 and soot. Only the 0D modelling of the flame dynamics is presented here, thus the awareness of the pollutants formation will not be detailed in this paper.

The developed 0D combustion model is based on the bond graph formalism[4] , allowing multi-physics simulations (e.g. hydraulic, mechanical, electrical, etc) and a network approach (i.e. a discretization of the fluid domain into several volumes). This allows to account for physical phenomena with spatial variations, such as heat transfer, dilution, turbulent flow field, etc. The premixed flame is entirely modelled with one single element.

2. Flow equations of the combustor element

This work is based on the 3D CFM model initially developed for RANS[5] and LES[15] calculations. This formalism is well adapted to premixed turbulent combustion processes with the assumption of a flamelet regime, which means that fuel oxidation is faster than turbulence characteristic time and hence the flame is thin in comparison with all scales of the turbulent flow. The CFM approach divides the gas into two zones : Unburnt Gases (UG) and Burnt Gases (BG) which are separated by the flame surface propagating from the burnt gases towards the fresh mixture, Figure 2. It can be noticed that post-flame chemistry occurs in the whole burnt gases zone, but this process will not be discussed in this paper.

The several equations governing this model were initially written for 3D simulations and then reduced by integrating them over the combustor volume, where only premixed combustion occurs, in order to obtain the 0D combustion model presented here. This reduction is based on several assumptions. First, it is assumed that UG and BG zones are completely described by their respective mass, volume, composition and temperature. Then, both of these zones shares the same pressure value and the gas is assumed as ideal. Finally, the UG and BG mixtures are supposed to be homogeneous in pressure, temperature and composition. This latter point added to the fact that the premixed combustion is entirely modelled with one single element involve that the flame parameters are constant along the entire flame front.

Simulating the temporal evolution of combustor features with the CFM formalism requires a sufficient amount of transported variables. These quantities are temporally transported by means of equations initially corresponding to a 3D case, such as mass and energy balance equations. Moreover, the presence of two distinguished zones (UG and BG) involves the transportation of additional variables. In this work, it has been chosen to transport species densities, ρ_i, and enthalpy, $H = mh$, of the UG zone (with UG exponent) and of the entire volume. The equality of the UG and BG pressure enables to transport the pressure, P, once with one perfect gas equation of state. To follow the evolution of the flame dynamics in transient conditions, the UG volume, V^{UG}, is transported, which is detailed in section 3.4.

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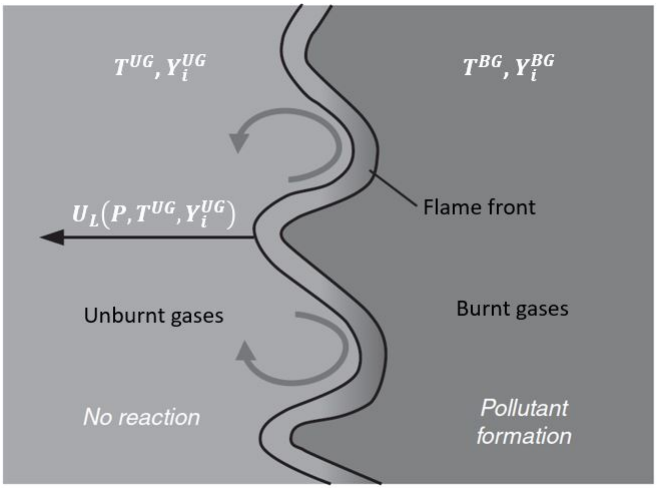

Figure 2: Visualisation of the UG and BG separation with the CFM approach. U_L stands for the laminar flame speed, T and P for the temperature and pressure and Y_i is the mass fraction of the species i.

Table 1: Summary of the transport equations associated with their corresponding variables.

Equations	Transported variable	Units
Mass balance in the global component	ρ_i	$[kg/m^3]$
Mass balance in the fresh gases zone	ρ_i^{UG}	$[kg/m^3]$
Energy balance in the global component	mh	$[J]$
Energy balance in the fresh gases zone	$(mh)^{UG}$	$[J]$
State equation	P	$[Pa]$
Flame dynamics	V^{UG}	$[m^3]$

2.1 Fresh and burnt gases

Within the combustor, the global mass variation of the species i is dependent on the intake and exhaust mass flow rates, respectively $\mathrm{d}m_i|_{in}$ and $\mathrm{d}m_i|_{out}$, and on the mass variation due to combustion, $\mathrm{d}m_i|_{comb}$:

$$\frac{\mathrm{d}m_i}{\mathrm{d}t} = \left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{in} - \left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{out} + \left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{comb} \tag{1}$$

where $\mathrm{d}m_i|_{comb}$ is positive for production of the species i and negative for consumption of the species i.

The global enthalpy, regrouping UG and BG, is transported via the equation below:

$$\frac{\mathrm{d}mh}{\mathrm{d}t} = V\frac{\mathrm{d}P}{\mathrm{d}t} - \frac{\mathrm{d}Q_{wall}}{\mathrm{d}t} + \frac{\mathrm{d}Q_{comb}}{\mathrm{d}t} + \left.\frac{\mathrm{d}mh}{\mathrm{d}t}\right|_{in} - \left.\frac{\mathrm{d}mh}{\mathrm{d}t}\right|_{out} \tag{2}$$

where $\mathrm{d}Q_{wall}$ is the heat losses of the system through the walls surrounding the flame tube and $\mathrm{d}Q_{comb}$ is the heat release of the combustion process, detailed in section 2.2.

The UG composition is dependent on the intake mass flow and on the mass flow rate crossing the flame front, $\mathrm{d}m_i|_{ff}$:

$$\frac{\mathrm{d}m_i^{UG}}{\mathrm{d}t} = \left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{in} - \left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{ff} \tag{3}$$

The UG temperature, T^{UG}, is obtained by inversion of the fresh gases enthalpy, h_{UG}, and accounting for the fresh gases composition.

$$\frac{\mathrm{d}(mh)^{UG}}{\mathrm{d}t} = V^{UG}\frac{\mathrm{d}P}{\mathrm{d}t} - \frac{\mathrm{d}Q^{UG}}{\mathrm{d}t} + \left.\frac{\mathrm{d}(mh)^{UG}}{\mathrm{d}t}\right|_{in} - \left.\frac{\mathrm{d}(mh)^{UG}}{\mathrm{d}t}\right|_{out} \tag{4}$$

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where Q^{UG} is the heat losses of the UG system towards BG and combustors walls. It is assumed to be negligible.

The BG composition is directly calculated from the global and the UG composition:

$$m_i^{BG} = m_i - m_i^{UG} \tag{5}$$

The BG temperature, T^{BG}, is also obtained by inversion of the burnt gases enthalpy by accounting for the BG composition, determined from the global enthalpy and the UG enthalpy:

$$h^{BG} = \frac{mh - (mh)^{UG}}{m^{BG}} \tag{6}$$

2.2 Heat release process

The chemical reactions of combustion are summarized by a single step mechanism, which is dependent on the fuel/air equivalence ratio in the fresh gases in order to well predict the CO formation produced through the flame front in rich premixed combustions. This two-in-one equation was proposed by Lafossas et al.[7] :

$$\begin{cases} C_xH_yO_z + \left(x + \frac{y}{4} - \frac{z}{2}\right) O_2 \rightarrow xCO_2 + \frac{y}{2}H_2O & ; \quad \phi < 0.98 \\ \\ \alpha_1 \left[C_xH_yO_z + \left(x + \frac{y}{4} - \frac{z}{2}\right) O_2 \rightarrow xCO_2 + \frac{y}{2}H_2O\right] + \\ (1-\alpha_1)\left[C_xH_yO_z + \frac{x-z}{2}O_2 \rightarrow xCO + \frac{y}{2}H_2\right] & ; \quad \phi \geq 0.98 \end{cases} \tag{7}$$

where the coefficient α_1 is used to reflect the dependency of the CO formation to the equivalence ratio, ϕ, which is defined as ratio of the actual fuel/air mass ratio to the stoichiometric fuel/air mass ratio:

$$\alpha_1 = \left(0.98\frac{4x + y - 2z}{\phi} - 2(x - z)\right) \Big/ (2x + y) \tag{8}$$

The heat released by the combustion process, $\mathrm{d}Q_{comb}$, is expressed as:

$$\frac{\mathrm{d}Q_{comb}}{\mathrm{d}t} = \sum_i h_{f,i} \left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{comb} \tag{9}$$

where $h_{f,i}$ is the mass enthalpy of formation of species i and $\mathrm{d}m_i|_{comb}$ is the mass variation of species i due to combustion, Equation (10). Moreover, according to the CFM formalism, the mass variation of species i due to combustion is directly correlated to the fuel mass consumption, $\mathrm{d}m_F|_{comb}$, and to stoichiometric coefficients of the fuel oxidation reactions, ν_i:

$$\left.\frac{\mathrm{d}m_i}{\mathrm{d}t}\right|_{comb} = \nu_i \frac{W_i}{W_F} \left.\frac{\mathrm{d}m_F}{\mathrm{d}t}\right|_{comb} \tag{10}$$

where W_i denotes the molecular weight of species i.

It can be noticed that the fuel mass consumption, $\mathrm{d}m_F|_{comb}$, corresponds to the fuel mass which burns during the flame front crossing. As a result, by adding the fuel part which does not burn during the flame front crossing (in very rich cases), the total UG fuel mass flow rate which crosses the flame front, $\dot{\Omega}$, can be expressed as:

$$\dot{\Omega} = \left.\frac{\mathrm{d}m_F}{\mathrm{d}t}\right|_{comb} + T_F^{UG \rightarrow BG} \tag{11}$$

where $T_F^{UG \rightarrow BG}$ is the fuel mass flow rate directly transferred to BG without burning for very rich flames.

3. Fuel consumption equation closure

The modelling of the flame dynamics is essential for this new 0D combustion model which aims to catch a realistic temporal evolution of combustion parameters in transient conditions. The following details how the CFM formalism is reduced from its 3D form to 0D equations.

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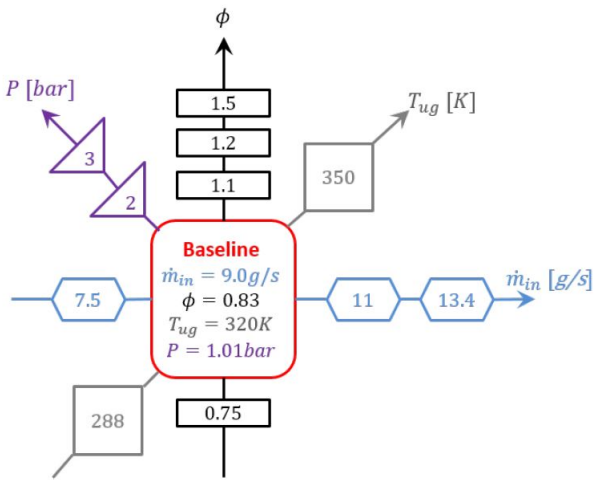

Figure 3: Visualisation of the different operating points simulated for the parametric study.

3.1 Numerical database generation

As shown in Equation (11), accounting for the unsteady flame behaviour during transient conditions requires a temporal derivative of the fuel mass crossing the flame front. This quantity can be obtained by using 3D simulation results. Accordingly, a parametric study of the PRECCINSTA burner with 3D simulations was carried out: twelve simulations were performed around the baseline case defined by a fuel/air mixture mass flow rate of $9g/s$, an equivalence ratio of 0.83, a UG temperature of $320K$ and an atmospheric pressure. Variations are operated on these four quantities with the aim to cover the largest range of operating points, Figure 3.

LES computations were performed with the code AVBP, co-developed by CERFACS and IFPEN, with gaseous species and gas assumed to be ideal. A 2 steps reduced chemical scheme for methane/air combustion is used, taking into account 6 species : CH_4, O_2, CO_2, CO, H_2O and N_2. The mesh contains 4.5 million cells and the Thickened Flame (TF) formalism is chosen[6] . This formalism consists in artificially thickening the flame front in order to properly discretize the flame front, while avoiding the need to refine the computational mesh. The TTGC numerical scheme is chosen.

The numerical setup is shown in Figure 4. A mass flow rate composed of air and methane is imposed on the inlet of a plenum placed upstream the swirler. A large plenum is placed downstream the exhaust in order to minimize the impact of the pressure condition imposed on the outlet. A mass flow rate boundary condition is introduced into the outlet plenum to force a co-flow constituting an additional precaution against spurious numerical effects. In addition, in order to properly account for the acoustic waves propagation and reflection, inlet and outlet boundary conditions are based on Navier-Stokes Characteristic Boundary Conditions (NSCBC)[12] . Finally, all the walls are assumed as no-slip adiabatic walls.

3.2 0D fuel consumption correction

A difficulty arises from the fact that the 0D model is based on the CFM formalism, Equation (23), whereas 3D simulations of gas turbines flames are usually based on the TF formalism, because of the presence of non premixed combustion.

The developed methodology consists of the use of 3D simulations results to close the 0D fuel consumption equation, but the point is that the main difference between these two formalisms is precisely the determination of the fuel consumption. Indeed, as it was previously seen, the CFM fuel consumption is linked to thermodynamics and turbulent flow, whereas the TF fuel consumption is directly calculated with chemical kinetics via Arrhenius laws. It leads to a different representation of the flame front and, therefore, the flame surface resolved with CFM is not equal to the resolved one with the TF model. Accordingly, some parameters of the 0D fuel consumption must be adapted to compensate these discrepancies.

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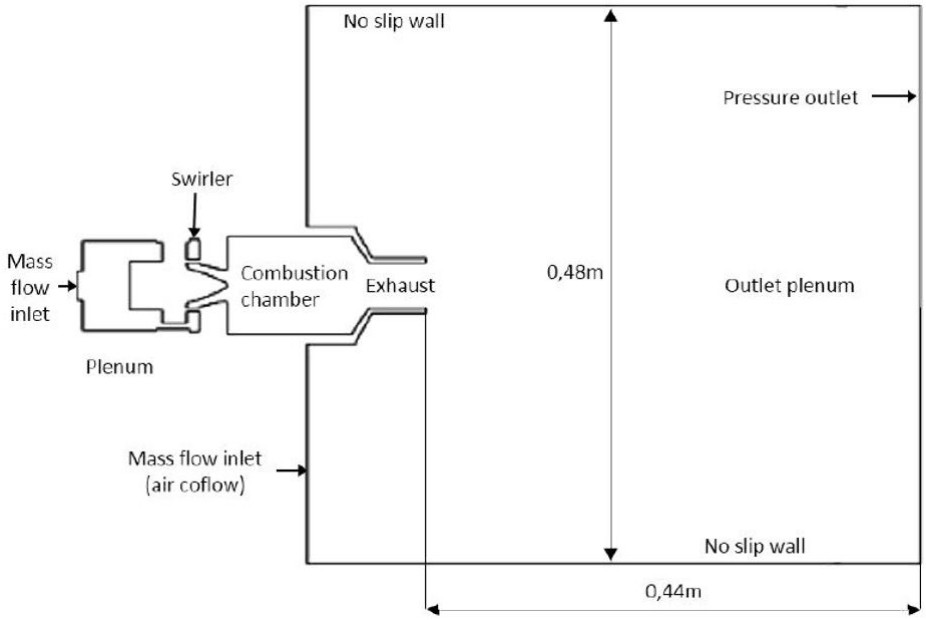

Figure 4: Schematic of the PRECCINSTA burner computational domain and boundary conditions.

3.3 3D CFM formalism reduction

From the 3D CFM formalism, the local fuel consumption, $\dot{\omega}_F$, is expressed as a function of the fuel mass density in UG, ρ_F^{UG}, the laminar flame speed, U_L and the flame surface density, $\overline{\Sigma}$.

$$\dot{\omega}_F = \rho_F^{UG} U_L \overline{\Sigma} \tag{12}$$

where the flame surface density is explicitly transported in the 3D CFM model.

The local fuel consumption can be spatially integrated over the whole premixed combustion domain in order to obtain the global UG fuel mass flow rate crossing the flame front:

$$\dot{\Omega} = \int \dot{\omega}_F \, \mathrm{d}V = \int \left(\rho_F^{UG} U_L \Sigma \right) \, \mathrm{d}V \tag{13}$$

Moreover, the flame surface density can also be expressed as:

$$\overline{\Sigma} = \overline{|\nabla c|} \tag{14}$$

where c is the combustion progress variable, which is equal to zero in UG and to unity in BG:

$$c = 1 - \frac{m^{UG}}{m} \tag{15}$$

This implies that c varies only through the combustion process, and $\overline{|\nabla c|}$ is accordingly different from zero within the flame front. Since the flame surface density differs from zero only in the flame area, the spatial integration can be limited to the flame front in Equation (13). Moreover, it was seen that accounting for the entire premixed flame in one single element and assuming a homogeneous mixture involve that the flame parameters, like the fuel mass density and the laminar flame speed, can be considered homogeneous along the flame front. Accordingly, the UG fuel consumption rate can be written as:

$$\dot{\Omega} = \rho_F^{UG} U_L \int \overline{\Sigma} \, \mathrm{d}V \tag{16}$$

This definition is very useful in the reduction process because the spatial integration of the flame surface density corresponds to the total flame surface:

$$S_{tot} = \int \overline{\Sigma} \, \mathrm{d}V \tag{17}$$

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Therefore, Equation (17) shows that the fuel consumption requires the total flame surface value, which cannot be directly computed in 0D simulations because of its high turbulence dependency, which is intrinsically space-dependent. The key of this reduced model is to use a temporally averaged flame surface, S_m, only dependent on the operating conditions, and then correct it by a wrinkling coefficient, Ξ.

First, a set of mean flame surface data can be obtained from multiple 3D simulations covering a wide range of operating conditions by using the following definition:

$$S_{mean} = \int |\nabla < \overline{c} > | \, \mathrm{d}V \tag{18}$$

where < .. > stands for a temporal average operator.

The flame surface wrinkling coefficient can also be obtained with 3D simulation. Indeed, it must be noticed that the whole flame surface is not fully resolved with 3D simulations because of the mesh size used with LES methods. Therefore, the flame surface density is decomposed into the resolved flame surface density, which is the gradient of the resolved combustion progress variable and a subgrid scale (sgs) wrinkling factor, Ξ_{sgs}, accounting for the unresolved flame surface:

$$\overline{\Sigma} = \Xi_{sgs} |\nabla \overline{c}| \tag{19}$$

Accordingly, assuming a Homogeneous and Isotropic Turbulence (HIT), the flame surface wrinkling is considered constant along the flame front. Therefore, the Equation (17) becomes:

$$S_{tot} = \int \Xi_{sgs} |\nabla \overline{c}| \, \mathrm{d}V = \Xi_{sgs} \int |\nabla \overline{c}| \, \mathrm{d}V = \Xi_{sgs} S_{res} \tag{20}$$

where S_{res} is the resolved flame surface.

In a similar way to Ξ_{sgs}, another wrinkling factor may allow to obtain the resolved flame surface from the mean flame surface with the assumption of an HIT:

$$S_{res} = \int \Xi_{res} |\nabla < \overline{c} > | \, \mathrm{d}V = \Xi_{res} \int |\nabla < \overline{c} > | \, \mathrm{d}V = \Xi_{res} S_{mean} \tag{21}$$

Therefore, a relation enables to link the mean and the total flame surface with a global wrinkling factor, Ξ:

$$S_{tot} = \frac{S_{tot}}{S_{res}} \frac{S_{res}}{S_{mean}} S_{mean} = \Xi_{sgs} \Xi_{res} S_{mean} = \Xi S_{mean} \tag{22}$$

These relations lead to the following writing of the 0D fuel consumption:

$$\dot{\Omega} = \rho_F^{UG} U_L \, \Xi \, S_{mean} \tag{23}$$

Finally, it can be noticed that the mass-weighted resolved progress variable, $\tilde{c}$, is more easily accessible from LES results. By neglecting the impact of the flame curvature, the following definitions of flame surfaces are retained:

$$S_{res} \approx \int |\nabla \tilde{c}| \, \mathrm{d}V \tag{24}$$

$$S_{mean} \approx \int |\nabla < \tilde{c} > | \, \mathrm{d}V \tag{25}$$

At this step, four quantities have to be modelled in order to resolve the 0D fuel consumption, whatever the operating condition. These quantities are the fuel mass density in UG, the laminar flame speed, the wrinkling coefficient and the mean flame surface.

3.4 Mean flame surface

The above mentioned parametric study revealed a correlation between the UG volume and the mean flame surface, which can be justified by the geometrical link connecting them. Indeed, an increase of the UG volume means that UG are less consumed, which involves that the mean flame surface extends far from the injector. As a result, for a given operating condition of the burner, the mean flame surface and the UG volume, normalised by LES results of a reference case (respectively A_{m_0} and V_0^{UG}), are correlated by a linear trend, Figure 5. It means that the UG volume evolves in a homothetic manner without shape change. This linear law is slightly modified when V^{UG} tends towards zero, in order

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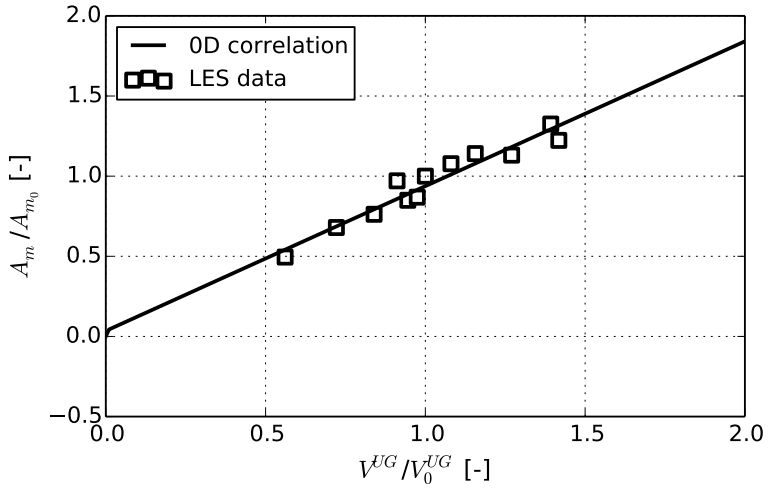

Figure 5: Variation of the normalised mean flame surface with respect to the normalised UG volume.

to avoid a positive surface flame value in case of absence of fresh gases. The used correlation is given by the following equation:

$$\frac{A_m}{A_{m_0}} = C_1 \frac{V^{UG}}{V_0^{UG}} + C_2 \left[1 - exp\left(-\frac{V^{UG}}{V_0^{UG}} \frac{1}{C_3}\right)\right] \tag{26}$$

where C_1, C_2 and C_3 are constants, respectively fixed to 0.9033, 0.0345 and 1.10^{-6}.

This correlation allows to extract the mean flame surface information from the UG volume, which can be transported in the 0D combustion model by temporally deriving the ideal gas law.

3.5 Fuel mass density in UG

Fuel mass density is directly transported in the 0D combustion model via the 0D mass balance equation conditioned to UG, see Table 1.

3.6 Laminar flame speed

Different empirical laminar flame speed correlations are given by Metghalchi et al.[10,11,13] depending on thermodynamic conditions and fuel properties. The 0D combustion model takes into account that the PRECCINSTA burner is fed with methane and air with the use of a methane flame speed correlation:

$$U_L(\phi, T^{UG}, P)\big|_{Metghalchi} = a_1 \left[1 + a_2(\phi - 1) + a_3(\phi - 1)^2\right] \left(\frac{T^{UG}}{T_{ref}^{UG}}\right)^{b_1} \left(\frac{P}{P_{ref}}\right)^{b_2} \tag{27}$$

The correlation coefficients values are given in Table 2:

Table 2: Coefficients values of the laminar flame speed correlation of a methane premixed flame.

Coefficient	$a_1[-]$	$a_2[-]$	$a_3[-]$	$b_1[-]$	$b_2[-]$	$T_{ref}^{UG}[K]$	$P_{ref}[Pa]$
Value	0.382	0.237	-3.411	1.854	-0.434	298	1.0^5

This empiric law is validated for the following range of fresh gases temperature, pressure and equivalence ratio :

$$298K < T^{UG} < 650K \quad ; \quad 1bar < P < 40bar \quad ; \quad 0.8 < \phi < 1.2 \tag{28}$$

Nevertheless, an extrapolation of the empiric law was made to extend the equivalence ratio range. In addition, as it was previously explained, a correction must be applied to the laminar flame speed correlation because the fuel

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consumption calculation methodology is different for the CFM and the TF formalisms. Indeed, whereas the laminar flame speed is directly calculated from Metchalchi correlation in CFM, a post-processing must be applied to obtained U_L in TF from the fuel consumption rate:

$$U_L^{TF} = \frac{\dot{\Omega}^{TF}}{\rho_F^{UG} S_{tot}} \tag{29}$$

However, this post-processing calculation leads to errors in the U_L^{TF} estimation. Accordingly, the correction coefficient is the ratio of the laminar flame speed calculated by Equation (29) to U_L calculated from the Metghalchi correlation at the baseline case:

$$U_L = \frac{U_{L,0}^{TF}}{U_{L,0}^{TF}(\phi, T^{UG}, P)\Big|_{Metghalchi}} U_L\left(\phi, T^{UG}, P\right)\Big|_{Metghalchi} \tag{30}$$

where $U_{L,0}^{TF}\left(\phi, T^{UG}, P\right)\Big|_{Metghalchi}$ is the laminar flame speed at baseline conditions calculated with the Metghalchi correlation using data extracted from 3D simulation baseline, and $U_{L,0}^{TF}$ is the laminar flame deduced from the TF fuel consumption rate, $\dot{\Omega}^{TF}$, also resolved at the baseline conditions.

3.7 Flame surface wrinkling

According to the CFM formalism, flame/turbulence interaction results in a wrinkled flame front propagating at laminar flame speed towards unburnt gases. The Bradley[3] correlation accounts for the effects of the aerodynamics via the ratio of the integral length scale, l_T, to the laminar flame thickness, δ_L, but also the effects of turbulence via the ratio of turbulent velocity fluctuations, u', to laminar flame speed. It can also be noticed that the effects of thermo-diffusive properties of the mixture are taken into account through the Lewis number, *Le*:

$$\Xi = 1 + \frac{0.95}{Le} K_{Brad} \sqrt{\frac{l_T}{\delta_L}\frac{u'}{U_L}} \tag{31}$$

where the coefficient K_{Brad} aims to compensate the Bradley correlation discrepancies and errors coming from the variables estimations. This coefficient is deduced from the LES parametric study, and its value is fixed at 0.2. The Lewis number is equal to unity in the performed simulations.

The laminar flame thickness is calculated by means of the Blint correlation given below:

$$\delta_L = \frac{2\lambda^{UG}}{\rho^{UG} c_p^{UG} U_L}\left(\frac{T^{BG}}{T^{UG}}\right)^{0.7} \tag{32}$$

where λ^{UG} is the thermal conductivity of the fresh gases mixture and c_p^{UG} is the fresh mixture specific heat at constant pressure.

The integral length scale is defined as the size of the biggest vortex within the turbulent flow. However, the 0D l_T modelling is complex. It is accordingly assumed that the integral scale is equal to the diameter of the flame at the anchoring point on the swirler nozzle, which is $7mm$.

A linear correlation between the turbulent velocity fluctuations in UG and the pressure drop in the combustion chamber, which are normalised by LES results of the baseline case (respectively $u_0'^{UG}$ and $(\Delta P/P)_0$), was observed for the PRECCINSTA burner. This linear correlation is compared to LES results in Figure 6 and defined by the following equation:

$$\frac{u'^{UG}}{u_0'^{UG}} = C_1 \left(\frac{\Delta P}{P}\right)\Big/\left(\frac{\Delta P}{P}\right)_0 + C_2 \tag{33}$$

where C_1 and C_2 are constants, respectively set to 0.56916 and 0.5646.

An aerodynamic pressure drop expression in non-reactive flow[8] is used:

$$\frac{\Delta P}{P} = \beta \frac{r^{UG}}{2}\left(\frac{\dot{m}\sqrt{T^{UG}}}{AP}\right)^2 \tag{34}$$

where β is a non-dimensional constant, r^{UG} is the perfect gas constant in the UG zone, $\dot{m}$ is the mass flow rate within the flame tube and A is a flame tube reference cross-section.

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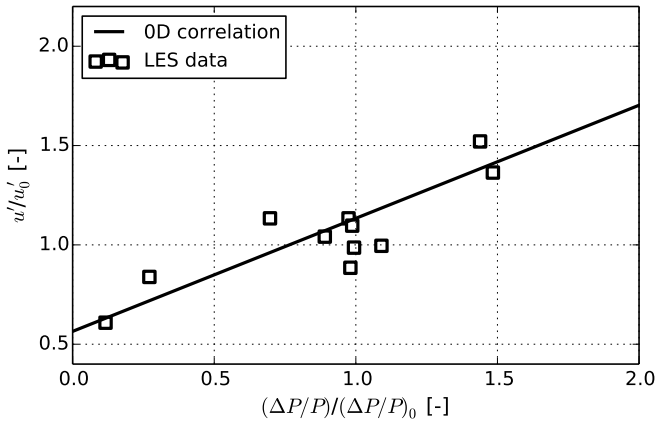

Figure 6: Evolution of the normalised turbulent velocity fluctuations with the normalised pressure drop. The normalisation is made with results from the LES baseline case.

4. Results and validation

The 0D combustion model is validated against the LES results of the PRECCINSTA burner hosting a methane/air turbulent premixed combustion. The inlet of the flame tube element is connected to a source injecting air and gaseous fuel described by a mass flow rate, a temperature and a fuel/air ratio. Its outlet ends with a convergent exhaust linked to an atmosphere pressure source. The system simulation is carried out using the LMS Imagine.Lab Amesim software.

First of all, steady state simulations are carried out in order to correctly validate 0D results with LES data. Then, unsteady simulations are performed enabling to highlight the robustness of the developed model to describe transient operating conditions.

4.1 Steady state simulations

In the following, a set of steady-flow simulations is carried out with an evolution of the UG equivalence ratio and mass flow rate in order to compare 0D results against the LES results obtained over the parametric study.

Figure 7 shows the response of several physical values to an equivalence ratio variation ranging from 0 to 2, while the UG temperature, the outlet pressure and the fuel/air mixture mass flow rate are kept constant, respectively at $320.5K$, $1atm$ and $9g/s$. First, these graphs stress the capability of the 0D combustion model to account for lean and rich quenching at equivalence ratios of around 0.5 and 1.6, which is coherent with methane/air flames theory. The laminar flame speed presents a parabolic shape, due to the correlation employed[13] , and reaches a peak around stoichiometric conditions in accordance with LES data. Turbulent velocity fluctuations increase with the equivalence ratio, which is due to the parameter r^{UG} impacted by the increase of fuel mass fraction in Equation (34). This graph shows a gap between LES data and 0D results, which suggests improvements of the model for u' will be necessary in the future. The mean flame surface and the UG volume evolutions illustrate that the flame is located near the injector near stoichiometric conditions, which is consistent when the flame comes close to quenching conditions. The flame surface wrinkling is almost equal to 4 when combustion occurs and it is equal to unity when the flame is quenched. The difference with LES results is simultaneously due to turbulent velocity fluctuations, laminar fame speed and flame thickness 0D values. The BG temperature profile reveals a good agreement with LES results. The UG fuel consumption rate is globally well predicted by Equation (23), even if an overestimation can be noticed for the richest LES case. This discrepancy is due to the fact that the mixture mass flow rate is lower at this operating point because of instabilities. As a result, LES underestimates the UG fuel consumption rate at this point. Finally, the BG carbon monoxide mass fraction does not seem to be well predicted by the 0D combustion model. This error could be explained by the fact that no post combustion chemistry is implemented at present. Hence, implementing this post flame chemistry should improve the CO formation prediction in the 0D combustion model. This topic will be investigated in next steps of the model development.

A similar study is performed with an evolution of the mass flow rate, of which results are concentrated in the Figure 8. The UG temperature, the outlet pressure and the equivalence ratio are kept constant, respectively at $320.5K$, $1atm$ and 0.83. The 0D laminar flame speed does not vary with the mass flow rate as presented in Figure 8. Indeed, accordingly to premixed flame theory, laminar flame speed is not impacted by a change of mass flow rate.

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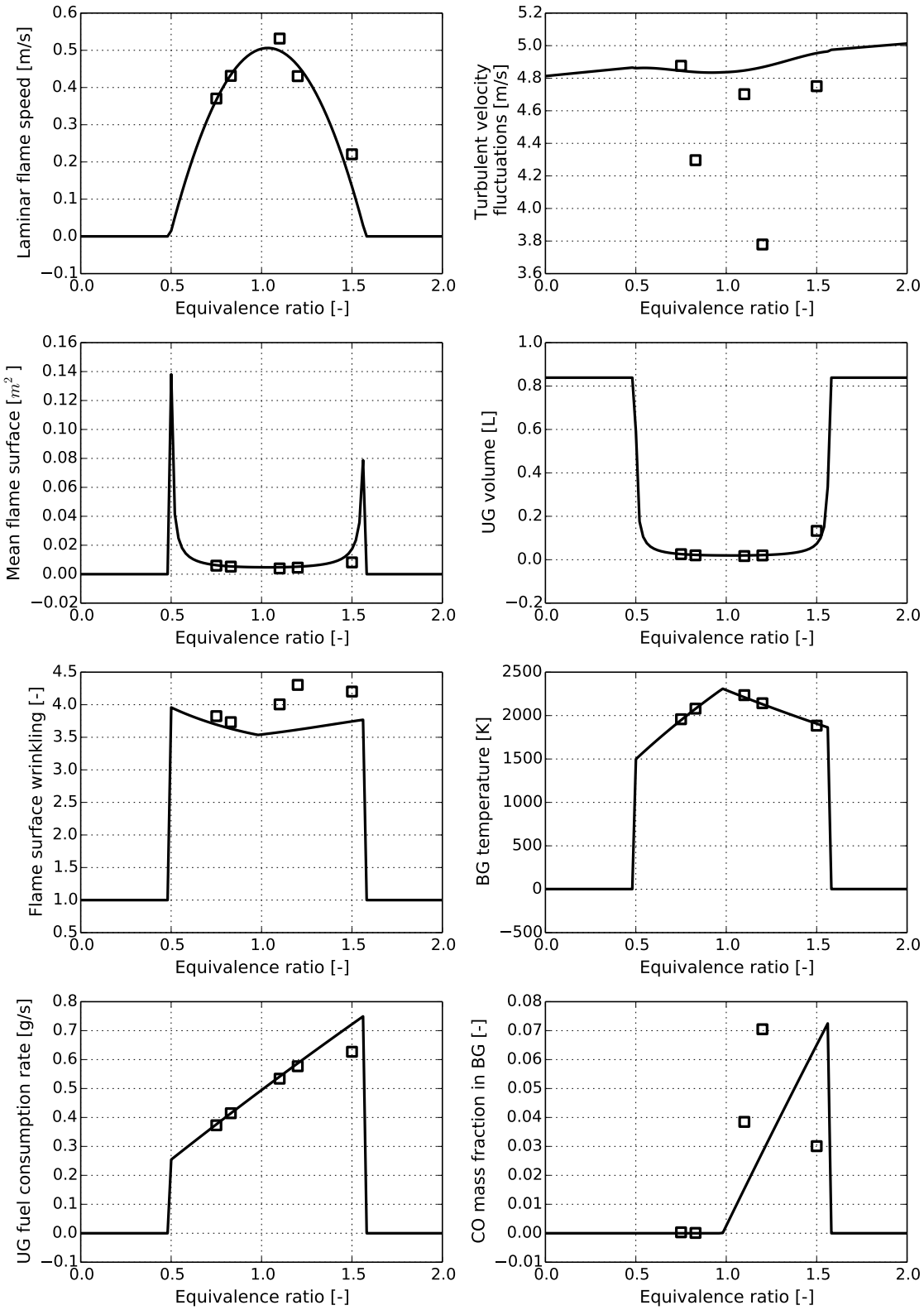

Figure 7: Validation of steady flow 0D simulations results (lines) against LES data (squares) with an equivalence ratio evolution.

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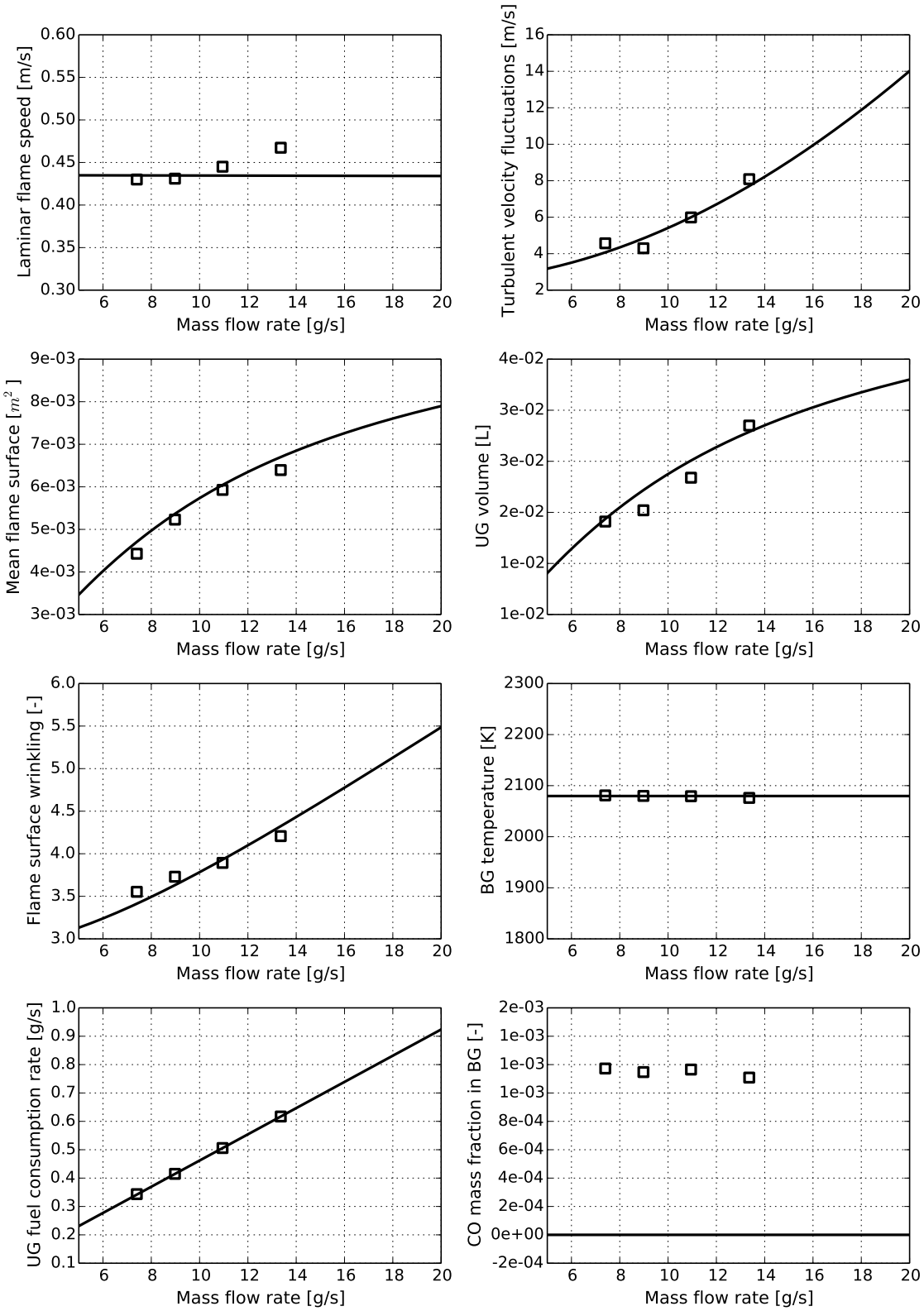

Figure 8: Validation of steady flow 0D simulations results (lines) against LES data (squares) with a mass flow rate evolution.

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The fluctuations observed in the LES results come from the use of Equation (29) to calculate U_L from TF combustion model. The turbulent velocity fluctuations increase with the mass flow rate, which impacts the flame surface wrinkling in the same way. However, the UG fuel consumption rate increase does not compensate the mass flow rate growth, which leads to a rise of the UG volume and of the mean flame surface as well. The BG temperature stays constant, in accordance with LES. Finally, the CO formation is not taken into account because the equivalence ratio is kept below 0.98, which does not lead to a production of CO according to the combustion representation of Equation (7).

4.2 Flame dynamics in transient conditions

The results presented above correspond to steady-state situations, and they do not represent the behaviour that can be observed in a combustion chamber. To highlight the capability of this 0D combustion model to account for transient operating conditions, simulation results as a function of time are presented below. In addition, this analysis allows to test the robustness of the model and to ensure good numerical stability for a wide range of situations.

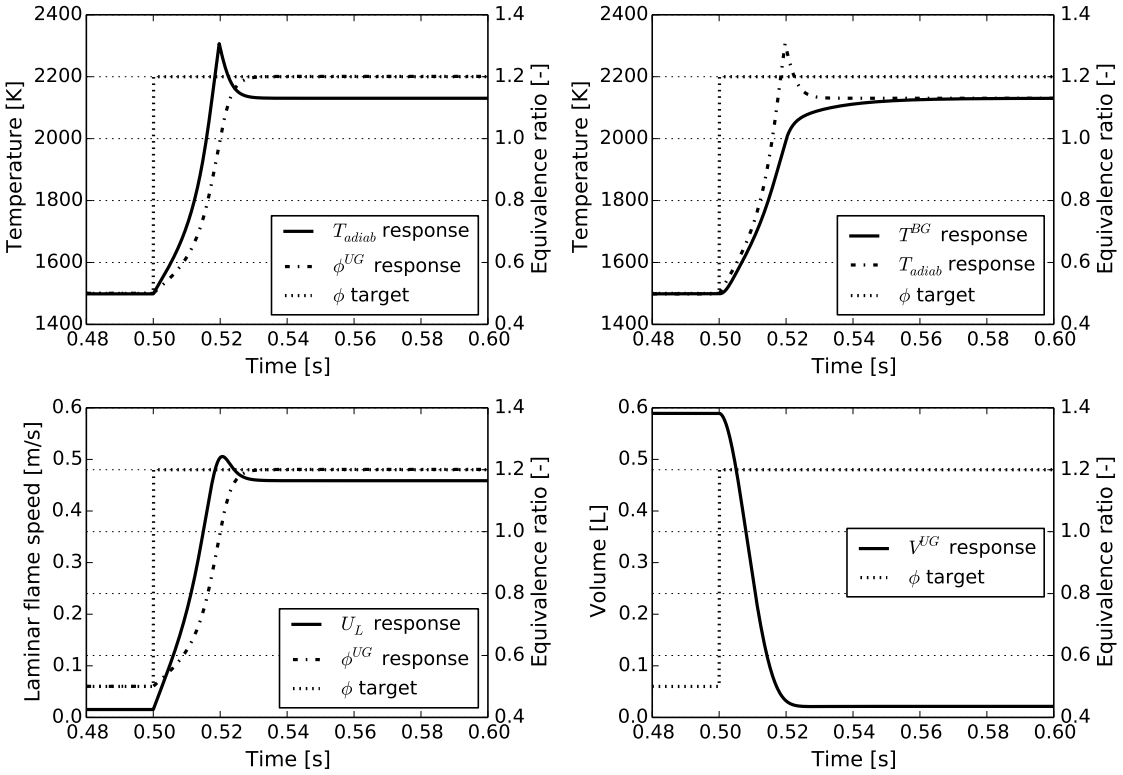

Figure 9: Response of the 0D combustion model to a suddenly increase of the equivalence ratio target.

In this study, the simulated combustor operates first at steady-state conditions, with a pressure value of $1atm$, a UG temperature of $320.5K$, a fuel/air mixture mass flow rate of $9g/s$ and an equivalence ratio of 0.5 to prevent the extinction of the flame. The target equivalence ratio is drastically increased when simulation time is equal to $0.5s$; this study could represent the realistic situation of a sudden increase of power demanded by the pilot of the aircraft. The change of boundary conditions from $\phi = 0.5$ to $\phi = 1.2$ is activated in $1.10^{-10}s$, as it is shown in Figure 9 by the dot line. The graph on the top left shows the inertia of the UG volume governed by a homogeneous mixture assumption, which implies that, even if the composition of the inlet mass flow rate drastically evolves, the whole zone composition response is submitted to a slight delay, which can be observed with the UG equivalence ratio response. However, the adiabatic temperature, which corresponds to the temperature of the gas in the near post-flame area, and the laminar flame speed plotted on the bottom left are directly linked to the UG equivalence ratio evolution and they evolve without inertia effect. Similarly to the UG equivalence ratio, the BG temperature plotted on the top right is subject to an inertia phenomenon, which leads to a slow evolution towards the equilibrium adiabatic temperature. Finally, with the dynamic evolution of the UG volume, this study highlights the 0D combustion model robustness, even in stiff transient conditions.

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5. Conclusion

In this paper, a new 0D combustion model was presented. The balance equations governing it were presented and the 0D description of the flame dynamics was detailed, from the reduction of 3D equations to the choice of parameters extracted from LES. At present, this 0D combustion model exclusively accounts for turbulent premixed combustion processes, using the CFM formalism with two zones separated by an infinitely thin flame surface. The flame propagation determines the UG fuel consumption rate, which depends on the UG composition, the laminar flame speed, the flame wrinkling and the mean flame surface. These quantities contribute to describe the flame dynamics and their determinations are carried out thanks to correlations closed with LES results of an academic premixed burners. The capability to use LES parametric studies to adapt 0D correlations to different architectures is an original methodology to increase the range of validity of this 0D combustion model to a high number of aeroengine combustor geometries. Finally, the 0D combustion model was validated against LES results of the PRECCINSTA burner and its robustness was highlighted in drastically stiff transient operating conditions.

However, some features must be developed to completely adapt this model to industrial gas turbines applications. First, the wrinkling correlation is not completely satisfactory with a constant integral scale. Therefore, a study concerning the turbulence and the vortex definition must be carried out. Then, the modelling of non-premixed combustion mode would allow to better represent the physical phenomena that occur inside the aeroengine flame tube. Finally, taking into account the post flame chemistry is the main way to well predict pollutants formation within an aeroengine.

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