On the Limits of Industrial Premixed Combustion Simulation Models

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Abstract

This work analyses the simulation potential of two premixed turbulent combustion models based on different combustion mechanism concepts: the Eddy Dissipation Concept based on the volume combustion mechanism, and the Turbulent Flame-speed Closure based on the thickened-wrinkled flamelets combustion mechanism.

Ability of simulating numerically a standard experimental test case (premixed methane-air combustion in a plane channel at high flow velocity) and the influence of flow parameters variation on the combustion process have been tested.

The paper shows that the flamelets model describes the standard experimental data more accurately. Furthermore, comparisons of the two models results obtained varying combustion flow parameters show the presence of quantitatively, and in one case even qualitatively different trends. These results are explained, and potentialities and limits of these models are discussed from an industrial premixed burner applications standpoint.

Nomenclature

c	progress variable	Y_i	mass fraction of species i
D	molecular diffusion coefficient m^2/s	χ	molecular heat transfer coefficient, m^2/s
G	stretch-factor	δ_l	laminar flame thickness, m
h_i	specific enthalpies, J/kg	δ_f	flamelet thickness, m
L	turbulent characteristic length, m	δ_t	turbulent flame zone width, m
$\dot{m}$	flow per unit time and unit area	ϵ	dissipation of turbulent kinetic energy, m^2/s^3
p	pressure, Pa	ϕ	equivalence ratio
T	temperature, K	η	Kolmogorov microscale, m
t	time, s	κ	turbulent kinetic energy, m^2/s^2
U_f	flamelet combustion velocity, m/s	μ	viscosity, kg /(m s)
U_l	laminar flame speed, m/s	ρ	mass density, kg/m^3
U_t	turbulent flame speed, m/s	τ_t	turbulent characteristic time scale, s
$\mathbf{u}$	velocity vector, m/s	τ_{ch}	chemical characteristic time scale, s
u'	turbulent RMS velocity, m/s	ν	kinematic viscosity, m^2/s
V	WSR reactor volume, m^3	α_t	turbulent Prandtl number
W_i	molecular weight of species i	$\tilde{}$	Favre average
w_i	chemical source term of species i	$\bar{}$	Reynolds average

Introduction

Premixed combustion technology is today one of the most promising ways to achieve reduction of pollutants emission from gas-turbine based power plants and future aeronautical engines. Therefore, there is an high interest in reliable models to simulate premixed turbulent combustion flows.

When dealing with laminar combustion, modeling is based on numerical solution of a set of equations including Navier Stokes equations, energy equation and chemical species transport equations. This system of equations, provided the suitable physical models, is in closed form, and can be solved with a degree of accuracy which depends both on the physical modeling sophistication and on the numerical approach used. Unfortunately, industrial combustion takes place in turbulent flows and simulation must deal with averaged equations which are not in closed form. Therefore, besides turbulence modeling, averaged equations need the so called "closures" for quantities generated by turbulent fluctuations and for mean reaction terms which, in particular, are made complex by the strong interaction existing between chemical reactions and turbulence.

The simplest turbulent combustion simulation approach is therefore to ignore completely the effects of pulsations of temperature and species concentration on the chemical reactions rates using a so called "quasi-laminar" combustion models. In this case, even using sophisticated models for turbulent transport, real tendencies of turbulent combustion cannot described quantitatively. Therefore, quantitatively correct numerical simulation of premixed turbulent combustion necessarily must be based on both a turbulence model and a a turbulent combustion model. Nowadays, several of such a models available in the literature [1] are already implemented in to research and commercial CFD codes.

It should be noted that many of the available premixed combustion models may reproduce realistically specific combustion flows; in fact, by "tuning" model parameters often satisfactory solutions can be obtained for *specific* test cases. But the question we would like to answer here is if these combustion models can be used for practical work, i.e. if they are really reliable for numerical simulation of combustion in real devices.

From our point of view, the most fruitful combustion models are those which not only make possible to reproduce quantitatively standard and classical experimental data, but also make possible to predict the influence of variations of operative conditions and geometry on the combustion process. In other words, they must contain the correct dependence on all turbulent and physico-chemical characteristics of the flow to grant correct results trends when varying combustion conditions (i.e., inlet velocities, turbulence intensity, mixture temperature and composition, mean pressure). Only at these conditions it can be possible to predict qualitatively the behaviour of the same machine for different operative condition, to improve the device or numerically look for an optimal design. Therefore, only combustion models that are based on realistic physical mechanisms can be the most efficient.

Turbulent combustion models for premixed combustion can be divided into two main groups: those which are based on volume combustion mechanism (instantaneous heat release is distributed in space similarly to combustion in well-stirred reactors) and those based on the flamelets combustion mechanism (instantaneous heat release takes place in thin flamelet sheets).

In this paper we discuss these important issues specifically related to correct simulation of premixed combustion in high turbulence flows that correspond to gas turbine combustion chambers. We compare results of numerical simulations for two turbulent premixed combustion models: the Eddy Dissipation Concept (EDC) [2, 3], based on a volume combustion mechanism, and the Turbulent Flame-speed Closure (TFC) [4, 5, 6, 7] based on the thickened and wrinkled flamelets combustion mechanism. These models are widely used [8, 9, 10, 11, 12] and are also available in commercial CFD packages (e.g. `FLUENT 6`). Here they have been applied to simulate a CH_4-air premixed combustion flow in an experimental plane combustor [13, 14]; results of these calculations will be here presented and commented.

Peculiarities of industrial premixed combustion

High intensity premixed industrial combustion takes place in a regime of developed turbulence with a continuous hierarchy of vortexes sizes from the largest, with size of the order of the device characteristic dimension, up to the smallest (Kolmogorov scale) which can be smaller than the laminar flame width ($\eta < \delta_l$).

There are two concept for premixed combustion modeling in flows with developed turbulence. The first one is based on the assumption that fine-scale turbulence extends the combustion zone, because the smallest vortexes, which are smaller than the expanded zone, are annexed in to the combustion zone. This leads to a flame width increase; this process leads to engulfement of more and more large vortices which ends up in a distributed combustion zone. The results is a “well stirred reactor” mechanism combustion, i.e., the flame heat release is distributed in space and time (volume combustion mechanism).

The second concept is based on the assumption that the process of engulfement of more and more large vortexes into the combustion zone has a natural limit. Therefore, the combustion mechanism leads to stationary flamelets of width larger than that of the laminar ones, taken in the same conditions, but smaller than the turbulence integral scale. This means that, in spite of the effect of the fine-scale turbulence, combustion take place in thickened and strongly wrinkled flamelet sheets (surface combustion mechanism).

Old theoretical estimations [5] and more recent experimental data [15] show that for real burners combustion regime the flamelet combustion mechanism with thickened and wrinkled flamelets [16] is the most probable.

For turbulent premixed combustion regimes, when $u' \gg U_f$, two peculiar phenomena can be underlined: the flame brush width increase and the weak turbulent flame velocity dependence on chemistry.

Increasing flame brush width

Premixed turbulent combustion takes place in a flame that is a turbulent mixing layer between cold unburned gas and hot products separated by a thin random flamelet sheet. The flame brush width is controlled mainly by turbulent diffusion as the transport of random flamelet sheets due to turbulent diffusion ($\sigma_t \approx (2D_t \cdot t)^{1/2} \sim (u'L \cdot t)^{1/2}$) is much more intense than the transport due to flamelet combustion velocity ($\sigma_c \sim U_f \cdot t$), i.e. $\sigma_t \gg \sigma_c$. Theoretically, after sufficient time, i.e. at sufficiently large distance from the beginning of flame is $\sigma_t \ll \sigma_c$ and the flame brush width becomes constant. Therefore, it is easy to deduce that, in terms of time, for $t < \tau_t(u'/U_f)^2$, where $\tau_t = L/u'$, we have flame brush with increasing width. In real combustion chambers a good estimate for the ratio u'/U_f can be of the order of 10, whereas for the residence time $t_r \sim 10\tau_t$. Therefore, in these devices we expect flame brushes with increasing width.

Dependence of the turbulent combustion velocity on chemistry

A further characteristic of turbulent premixed combustion in the flamelet combustion regime is the relatively weak dependence of the combustion velocity U_t on chemical kinetics with respect of what occurs for laminar premixed flames. This can be seen looking at how the U_t dependence on fresh mixture temperature ($U_t \sim T^{(0.4-0.5)}$, Doroshenko, 1956) is much weaker than that of U_l ($\sim T^2$)

The physical reason for this is a "hydro-dynamical compensation mechanism": increasing temperature, and then chemical rates, the increase of the flamelet velocity leads to a smoothing of the turbulent flame sheet and therefore to a reduction of the flamelet sheet area. Looking at the expression of the turbulent combustion velocity [5]: $U_t = U_f\overline{(\delta S/\delta S_0)}$ where δS is the area of a flame surface element and δS_0 is its projection normal to the direction of average flame front propagation, we see that the increase of U_f due to a faster chemistry is balanced by a decrease of the area ratio term. So the effect of temperature on U_f are damped by flame sheet area reduction leading to moderate effects on U_t and practically do not affecting the flame brush width.

It has to be emphasised that the hydrodynamical compensation mechanism for flows with strong turbulence ($u' \gg U_f, U_t$), leads to a weak dependence on chemistry only for flames with increasing flame brush width. For 1D stationary flames it is $U_t \approx u'$ i.e., a complete compensation occurs and $\overline{(\delta S/\delta S_0)} \sim (u'/U_f)$.

Therefore, a model which wants simulate effectively premixed combustion flames in the condition addressed by this work, must contain the flame brush increase behaviour and must account for the hydro-dynamical compensation mechanism. This can be hardly done directly in terms of modeling equations but more likely throughout some physical model which accounts for the effect these phenomena have on the combustion process.

Turbulent Flame-speed Closure (TFC Combustion Model)

The Turbulent Flame-speed Closure model (TFC) [4, 5, 6, 7] is an asymptotic flamelet combustion model valid at large turbulent Reynolds ($Re_t = u'L/\nu$) and Damköhler ($Da = \tau_t/\tau_{ch}$) numbers. It is based on the thickened and strongly wrinkled flamelet model with $U_f > U_l$ and $\delta_f > \delta_l$. This physical condition is achieved when $Da \gg 1$; $Re_t \gg 1$ and $Da^{3/2}\, Re_t^{-3/4} \ll 1 \ll Da^{1/2}$ where, in this latter expression, the left inequality is the condition for flamelet thickening and the right inequality is the condition for strong flamelet wrinkling.

The TFC model, besides averaged Navier-Stokes and turbulence model equations, requires to solve a further equation for the progress variable $\widetilde{c}$ ($\widetilde{c} = 0 \Rightarrow 100\,\%$ reactants; $\widetilde{c} = 1 \Rightarrow 100\,\%$ products):

$$\partial\,(\overline{\rho}\,\widetilde{c})/\partial t + \nabla\cdot(\overline{\rho}\,\widetilde{c}\,\widetilde{\mathbf{u}}) = \nabla\cdot[(\overline{\rho}\nu_t/\alpha_t)\;\nabla\,\widetilde{c}] + \rho_u\,U_t\,|\,\nabla\widetilde{c}\,|\,G \tag{1}$$

where the subscript u refers to unburned mixture, and the theoretical expression of the turbulent combustion velocity U_t reads:

$$U_t = A\,u'^{3/4}\,U_l^{1/2}\,\chi_u^{-1/4}\,L^{1/4} \tag{2}$$

Density and temperature can be evaluated according to weighted averages:

$$\overline{\rho} = [\,(1-\widetilde{c})/\rho_u + \widetilde{c}/\rho_b\,]^{-1} \qquad \overline{T} = (1-\widetilde{c})\,T_u + \widetilde{c}\,T_b \tag{3}$$

where the suffix b refers to burned mixture.

In Eq. (2) A is a empirical parameter which has been set $A = 0.51$ after experimental tests for spherical flames of several hydrocarbon fuels (CH_4, C_2H_6, C_3H_8) in bombs with artificial turbulators (for H_2 flames a better agreement with experiments has been found setting $A \simeq 0.6$) [5]. We want to stress that the exponents in the U_t expression being theoretically derived, do not contain any empirical setting [5]. It is such "rigid" theoretical construction which reduces to a minimum the number of empirical parameters and ensures a fixed dependence on physical controlling factors.

The stretch-factor G in Eq. (1) has been introduced in the combustion model to describe the bending of the turbulent combustion velocity due to flamelets extinction at increasing turbulence. In the test case simulated in this work we assumed this effect as negligible, and we put $G = 1$. Other test cases where it has been assumed $G < 1$ can be found in [6, 7].

This model is suitable for flames with increasing flame brush width, i.e. when $u' \gg U_f$ and for times $\tau_t < t < \tau_t(u'/U_f)^2$. Furthermore, it holds with the assumption of constant turbulent combustion velocity U_t, which means that the inclination of the flame brush with respect to the incoming fresh mixture is the same along the flame. From a physical point of view, this means that the model assumes the existence of a equilibrium small-scale structure of the random flamelet sheet which controls its area.

For times $t < \tau_t$ the model is not valid just because this times correspond to the formation of the wrinkled flamelet sheet and therefore its area is increasing in time, i.e. during this time U_t depends not only on mixture physico-chemical properties and on turbulence characteristics but also directly on time. This effect is important for SI engines (e.g. delay of ignition, that has physical turbulent nature) but it probably can be ignored when dealing with gas turbines combustion chambers. In our simulations we assumed that Eq. (1) is valid for $0 < t < \tau_t(u'/U_f)^2$.

When $t > \tau_t(u'/U_f)^2$ flamelet sheet transport due to flamelets combustion velocity would be comparable with its transport due to turbulent pulsation and for $t \gg \tau_t(u'/U_f)^2$ there will be flames with constant brush width. Our model does not describe such kind of flames, whose turbulent combustion velocity does not depend on chemistry ($U_t \sim u'$ [17]), but, as already underlined, we believe that this regime is not realized in industrial high intensity combustors where the residence time is shorter than $\tau_t(u'/U_f)^2$.

Finally we want to point out that this combustion model accounts for the compensation mechanism leading to a weaker dependence of U_t on chemical kinetics ($\sim (\tau_{ch})^{1/4}$) whereas for laminar combustion it is $U_l \sim (\tau_{ch})^{1/2}$

Eddy Dissipation Concept (EDC Combustion model)

The Eddy Dissipation Concept (EDC) of Magnussen [2] is a turbulent combustion model which gives an expression for the chemical species source terms accounting for the interaction between chemistry and turbulence in flames. It is based on physical modeling of turbulent processes which are relevant to turbulent combustion [3].

The main idea is that, provided a sufficiently high temperature, the chemical species react when are mixed at molecular scale. Therefore reactions can be assumed to take place within turbulent fine structures which, in this way, can be assumed to be like chemical reactors in which conversion from reactants to products is controlled by chemical rates and not by mixing of reactants (i.e., Well Stirred Reactors (WSR)). In these reactors the residence time depends on characteristic times of large (integral) and small (Kolmogorov) scale turbulence of the flow.

The EDC model, besides averaged Navier-Stokes and turbulence model equations, requires to solve for, ns chemical species and ne elements, $ns - ne - 1$ transport equations:

$$\partial(\overline{\rho}\,\widetilde{Y_i})/\partial t + \nabla\cdot(\overline{\rho}\,\tilde{\mathbf{u}}\,\widetilde{Y_i}) = \nabla\cdot[\overline{\rho}(\mathcal{D} + \nu_t/\alpha_t)\nabla\widetilde{Y_i}] + W_i\,\widetilde{\dot{w}_i} \quad i = 1,...,ns - ne - 1 \qquad (4)$$

where the mean source terms are modeled as [18]:

$$\widetilde{\dot{w}_i} = [\dot{m}^*\gamma^*\chi/(1 - \chi\gamma^*)]\left(Y_i^* - \widetilde{Y_i}\right)\rho^*/W_i \qquad (5)$$

The mass fraction of fine structures reads [2]:

$$\gamma^* = 9.7\cdot(\nu\epsilon/\kappa^2)^{3/4} \qquad (6)$$

and the fraction of fine structures which may react is assumed to be $\chi = 1$. The residence time in the fine structures is defined as:

$$\tau^* = 1/\dot{m}^* \equiv 0.411\cdot(\nu/\epsilon)^{1/2} \qquad (7)$$

The assumption that fine structures are like WSRs makes possible to obtain the fine structures composition, Y_i^*, via the solution of the WSR mathematical model , that is, a system of ODE which includes $ns - 1$ chemical species conservation equations plus the energy equation [19]:

$$\dot{m}(Y_i - Y_i^\circ) - \dot{\omega}_i W_i V = 0 \qquad (8)$$

$$\dot{m} \sum_{i=1}^{ns} (Y_i h_i - Y_i^\circ h_i^\circ) + Q = 0 \qquad (9)$$

where, besides the quantities already known, $\dot{\omega}_i$ is the molar rate of production by chemical reaction of species i per unit volume, and Q is the reactor heat loss. Superscript $^\circ$ indicates reactor inlet quantities. In this work we assumed no radiative losses for the reactors therefore $Q = 0$ and the energy equation reduces to the equality between reactant and products enthalpy. Once mean mass fractions are known, we calculate temperature from enthalpy and mean density from state equation.

Test Case Description: Experimental Premixed Combustor

The test case chosen to demonstrate the effectiveness of our study is an experimental combustor of simple geometry for which a large set of experimental data exists. This experimental work has been done by Moreau et al. [13, 14].

The experimental combustor is a plane channel feed with two flows: a homogeneous CH_4-air unburnt mixture and a burnt CH_4-air mixture. The combustor section is a square with a .10 m edge and it is 1.3 m long. The two inlet flows are divided by a splitter plate: the burned gas inlet is .02 m high, the fresh mixture inlet .08 m high. The chamber walls are uncooled and equipped with probes to measure temperature and chemical species concentrations during the test. Each test lasts for 30 seconds.

The burned gas entering from the lower inlet ignites the fresh mixture and stabilises the flame which then propagates along the combustor.

The combustor has been discretized using a Cartesian grid with 108×42 nodes. The two inlet flows are set up using constant profiles for velocity and other scalar quantities. Walls are assumed to be adiabatic and "zero gradient" conditions are imposed at outlet.

CFD Solver

The turbulent combustion models are implemented within the TANIT software [20] that is a parallel CFD code for gas turbines applications. The computational model is based on the solutions of the reactive Navier-Stokes equations for turbulent flows both in premixed and non premixed regime.

Turbulent transport is modeled via the Prandtl-Boussinesq hypothesis plus the $k - \epsilon$ model for the eddy viscosity. The chemical species diffusivities are set all equal to the same value and the Lewis number is assumed to be one ($Le = \lambda/(D\rho c_p) = 1$). Radiation has not been taken into account.

Using the Fractional Step Method solver, a preliminary velocity field is obtained solving momentum equation. This velocity is corrected by using the pressure field calculated from a Poisson equation obtained by the continuity equation that acts as a constraint condition to impose mass conservation. After updating the pressure and velocity field, the integration of scalars (i.e., turbulent kinetic energy κ, its dissipation ϵ, and thermochemical quantities) is performed. An upwind Finite Volume scheme is employed for the spatial discretization. The code has been extensively validated for reacting and non-reacting flows [20], [21].

Results

The test case has been simulated using both models with the same inlet conditions which are shown in Table 1. For the burned gas inlet, equilibrium composition has been assumed. Inlet conditions have been set up on the basis of the data found in [13] plus an initial tuning to reproduce profiles of velocity and turbulence intensity in the section nearer inlets. In fact, having only data on RMS velocity, we set up turbulence integral scales values to obtain the best agreement with experiments. The EDC has been run using the Chang and Chen 6-steps reduced mechanism for lean CH_4−air combustion which includes 10 chemical species [22].

Figure 1, which reports contour plots of the progress variable $\tilde{c}$ shows that the two models give qualitatively different structure of the flame brush: the flame brush width increases for the TFC model and remains nearly constant for the EDC model.

Running the two model which these boundary conditions the TFC produced almost immediately results in good agreement with experiments whereas EDC gave results strongly far from experimental data (see top of Fig. 2). Therefore to achieve the best performance EDC has requested a further model tuning which consisted in increasing the mass fraction of the burning structures. A comparison of those results with experiments can be found on the bottom of Fig. 2 which shows that both models give mean axial velocity in very good agreement with experiments for all the monitored combustor sections even thought the mean velocity peak is slightly overestimated. For this test case, both models reproduce fairly correctly the volume of gas burned by the premixed flame, i.e. the flame inclination in the channel. When looking at the concentration profiles reported in Fig. 3, we see that the TFC describes correctly experimental CH_4 concentration profiles and reproduces the flame width increase along the channel. On the contrary, EDC CH_4 concentration plots maintain the same slope for all the three sections showing that this model does not reproduces this characteristic behaviour of turbulent premixed flames. Similar results have been found for other chemical species profiles (O_2, CO_2) and for the temperature (non shown here).

The physical reason of these discrepancies are the different base concepts of the two models: for the TFC the flamelets combustion mechanism describes the space distribution of combustion as controlled by random movement of flamelets due to turbulent pulsations, whereas for the EDC the volume combustion mechanism describes the space distribution of combustion as controlled by turbulent transport and chemical kinetics. Therefore, the volume mechanism, i.e. the EDC model, produces a too steep fuel consumption leading to a thinner flame, and to concentration profiles which maintain the same slope for all the three sections, giving a flame of structure similar to that of a 1-D stationary flame.

Combustion Trends Analysis

For industrial application of combustion models we consider fundamental their ability of producing quantitatively accurate results for combustion at different initial temperature, pressures, velocities and flow turbulence. Only models which predict correctly these trend can be really considered useful for such industrial applications.

To observe this behaviour for the two turbulent combustion models presented here, we performed a parametric study: pressure, gas mixture inlet temperature, and turbulence intensity have been varied in a suitable range. The flame position has been used to trace models behaviour. This position has been assumed to be represented by the ordinate where the progress variable is $c = 0.5$ and as been indicated using a nondimensional coordinate

Y_0/h where $h = 0.02m$ is the height of the burned gas inlet. Results of this study are shown in Figs. 4-6.

Figure 4 shows that the pressure variations influence is qualitatively opposite for these two models. The decreasing of the turbulent combustion velocity for the TFC model is connected with the flamelets combustion mechanism: increase of pressure decreases laminar flamelets combustion velocity due to the decreasing of the molecular transfer coefficient. For the EDC the pressure increases raises the turbulent combustion velocity due to the increase of chemical reactions rates while turbulent transport remains nearly unchanged (the EDC is indifferent to molecular diffusion reduction).

The inlet mixture temperature influence on turbulent combustion velocity is shown in Fig. 5. It is much stronger for EDC than for TFC model: in fact for the EDC this dependence corresponds to that of a quasi-laminar model.

Dependence of turbulent combustion velocity on turbulence intensity for the TFC model is $U_t \sim u'^{3/4}$, while for EDC model it close to $U_t \sim u'^{1/2}$. Figure 6 shows this different behaviour.

We emphasise that for flamelets combustion mechanism, chemistry cannot be introduced in the model averaged equation in a direct way, but only through some physical model for flamelet and flamelet sheet. So the TFC flamelets model uses chemical kinetics in preprocessing to define U_l and does not requires to introduce this computationally costly process at running time. This fact plays in favour of the CPU time consumption; in fact the TFC model requires approximately fifteen times less computer time in comparison with the EDC+WSR combustion model (with 6 chemical kinetics steps, and 10 chemical species).

Conclusions

We analysed two premixed combustion models: the TFC based on the flamelets combustion mechanism and the EDC based on the volume combustion mechanism. Both these models are widely used to simulate premixed combustion flows. Experiments show that industrial premixed combustion flows have a flamelets mechanism (the subject still under discussion is the actual structure of the flamelets: laminar normal or narrowed by stretch or, on the contrary, thickened by the fine-scale turbulence).

For this reason, we think that in the test case studied here, even thought the "tuned" EDC model gave turbulent combustion velocity (i.e., flame inclination) in good agreement with experiments, it couldn't describe correctly the flame brush width. Instead, this can be done by using the TFC model. Furthermore, we have seen that EDC and TFC combustion models give different trends for parameter variations as they are based on different physico-chemical mechanisms.

It is also obvious that not every flamelets combustion model can give correct trends, i.e. can be used for industrial application. We think that preference must be given to models that are based on the physical mechanisms involved and which give the best agreement with proper standard experimental data. Being the TFC one of those, we would like to propose it as a possible candidate. Nevertheless more test on its validity and effectiveness are still necessary.

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Table 1: Inlet flow conditions used for the CFD simulations (S.I. units).

	u	u'	κ	$L_{(TFC)}$	$L_{(EDC)}$	ρ	T	ϕ
burnt mixture	108	23	793.0	1.3E-3	2.0E-3	0.1507	2240	0.87
unburnt mixture	65	8	100.0	4.4E-3	8.0E-3	0.573	600	0.87

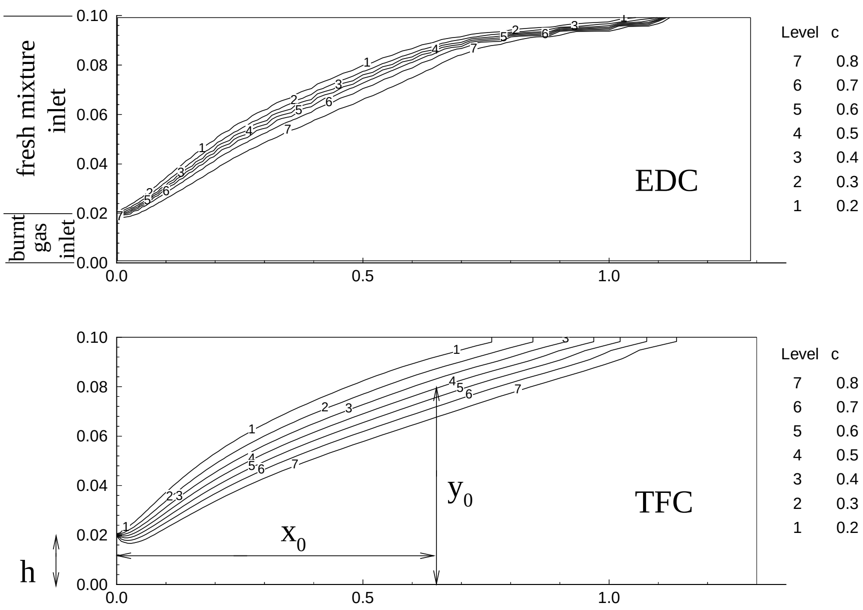

Figure 1: Progress variable contour plots (notice that the ratio between abscisses and ordinates scales is 0.2).

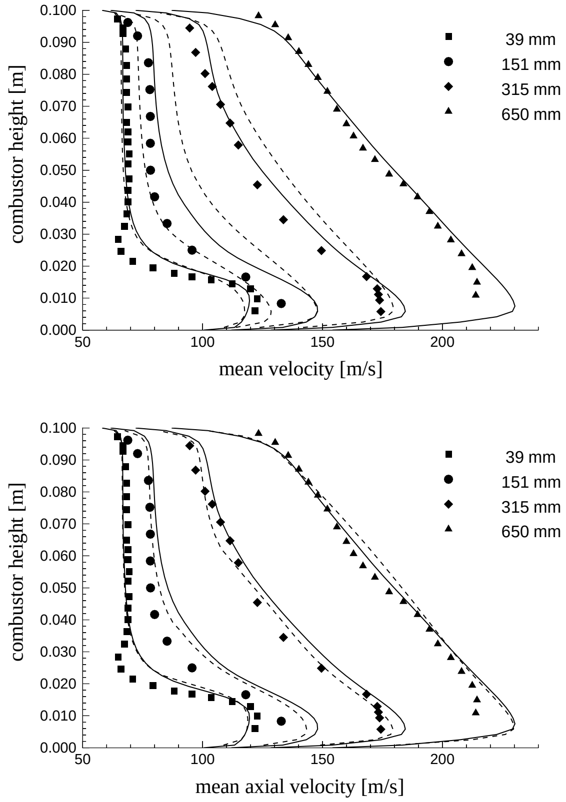

Figure 2: Mean velocity plots at different combustor sections: top EDC non tuned; bottom EDC tuned. EDC–dashed lines; TFC–solid lines; experiments–symbols.

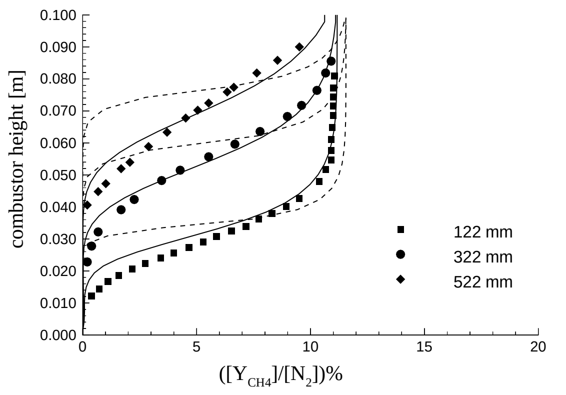

Figure 3: Mean CH_4 concentration plots at different combustion sections: EDC dashed lines; TFC solid lines; and experiments symbols.

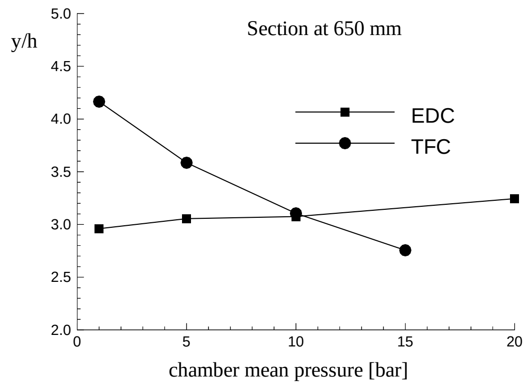

Figure 4: Pressure chamber effect on flame brush position at x_0 = .65 m from inlets (h = .02 m).

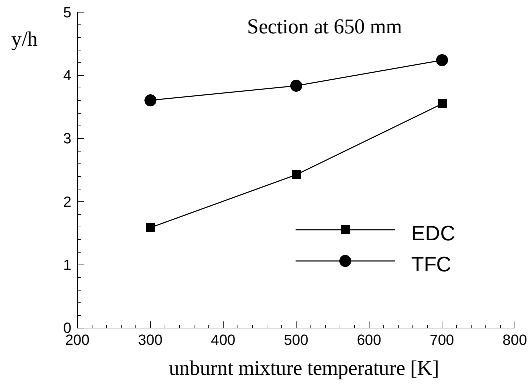

Figure 5: Combustible mixture temperature effect on flame brush position at x_0 = .65 m from inlets (h = .02 m).

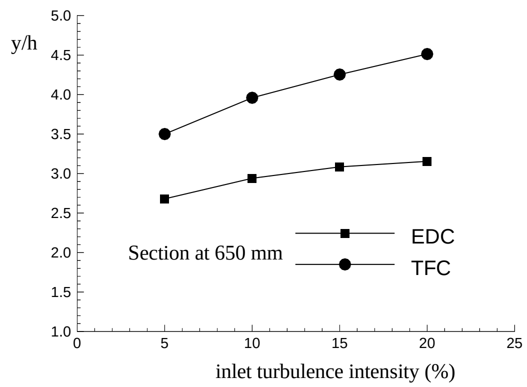

Figure 6: Inflow turbulent intensity effect on flame brush position at x_0 = .65 m from inlets (h = .02 m).