Simulations of Premixed Turbulent Stagnation Flames with a Flame Speed Closure Model

A. Lipatnikov and J. Chomiak

Department of Thermo- and Fluid Dynamics,
Chalmers University of Technology, Gothenburg, 41296, Sweden
e-mail: lipatn@tfd.chalmers.se

Abstract

Numerical simulations of premixed turbulent stagnation flames have been performed using the Flame Speed Closure (FSC) model in order: (1) to further test the model under inauspicious conditions, and (2) to test whether or not stagnation flames are equivalent to fully developed flames in homogeneous flow fields. The results indicate that the model well predicts the stationarity of these flames and their mean thickness. This suggests that the flames studied are not fully developed, because the FSC model cannot describe fully developed flames, in principle.

Introduction

In recent years, the so-called Flame Speed Closure (FSC) model, discussed, in detail, elsewhere [1, 2, 3], has been successfully applied to predicting the basic characteristics of various laboratory [2, 3, 4, 5, 6, 7, 8] and industrial [5, 9] premixed turbulent flames. Despite this wide validation of the model, certain authors [10] question it by highlighting that the FSC model cannot describe fully developed flames, in particular, planar flames of a constant thickness, which propagate at a constant speed against a stationary and spatially uniform unburned mixture flow. Indeed, it can easily be shown [3] that the following basic equation of the model

$$\frac{\partial}{\partial t}\left(\bar{\rho}\tilde{c}\right) + \frac{\partial}{\partial x_j}\left(\bar{\rho}\tilde{u}_j\tilde{c}\right) = \frac{\partial}{\partial x_j}\left(\bar{\rho}D_t\frac{\partial \tilde{c}}{\partial x_j}\right) + A\rho_u u' Da^{1/4} \mid \nabla\tilde{c} \mid, \tag{1}$$

predicts permanent growth of the mean flame thickness, δ_t, in the planar one-dimensional case. Here, c is the progress variable, x_j and u_j are the coordinates and flow velocity components, respectively; ρ is the gas density; the subscripts u and b label the unburned gas and products, respectively; $Da = \tau_t/\tau_c$ is the Damköhler number; $\tau_t = L/u'$ and $\tau_c = \kappa_u/S_L^2$ are the turbulent and chemical time scales, respectively; u', L, and D_t are the r.m.s. turbulent velocity, integral length scale, and diffusivity, respectively; κ_u is the heat diffusivity of the unburned mixture; S_L is the laminar flame speed; and A is a constant of the model. Both the Reynolds averages denoted by overbars and the Favre averages, such as $\bar{\rho}\tilde{c} = \overline{\rho c}$ and $c'' = c - \tilde{c}$, are used in Eq. 1.

Contrary to Peters [10], we do not think that the discussed property is a substantial drawback. From our point of view, a good model must describe real processes, rather than hypothetical ones. Numerous experimental data analyzed by us [11] indicate that many laboratory flames (spherical, Bunsen-type, V-shaped, etc.) are developing flames, rather than fully developed ones. For these reasons, many popular models, which deal with a hypothetical, fully developed regime of turbulent combustion, fail to predict the basic features of real flames, as discussed, in detail, elsewhere [11, 12]; whereas the FSC model is capable of doing so.

Nevertheless, the inability of Eq. 1 to yield a fully developed flame restricts the domain of applicability of the FSC model. Thus, the following issues appear to be of interest: Is this restriction important? Do fully developed laboratory premixed turbulent flames, which the FSC model cannot be applied to, exist? Premixed flames in stagnating turbulence seem to be a challenge to the FSC model from this standpoint. On the face of it, such planar and steady flames look like fully developed ones and many authors appear to share this opinion. However, this association may be questioned because stagnating flows are substantially spatially non-uniform, whereas fully developed flames must be steady and planar if the unburned mixture flow is stationary and spatially uniform.

The above reasoning explains the goal of this study, that is to simulate stagnation turbulent flames using the FSC model. Such studies offer the opportunity: (1) to further test the model under inauspicious conditions, and (2) to get insight into the relation between stagnation turbulent and fully developed flames. Indeed, if stagnation turbulent flames were equivalent to fully developed flames, Eq. 1 would fail to describe them. To the contrary, if Eq. 1 is able to describe stagnation turbulent flames, this would suggest that they are developing flames, rather than fully developed ones.

A numerical model

Premixed turbulent stagnation flames have been intensively investigated over the past 15 years. The state of the art of this problem was discussed in a recent paper by Bray et al. [13], in which the key references may be found. For this reason, we will restrict the discussion only to issues related directly to our goals.

In Ref. [13], a quite sophisticated submodel of the effects of heat release on turbulent transport has been developed; but, when validating it against experimental data, the authors used the measured profiles of the progress variable, rather than predicting them. We looked into the problem from another approach: The focus was placed on combustion modeling, whereas the turbulence submodel was simplified as much as possible; i.e., we simulated a flame stabilized in a hypothetical stagnating-like mean flow with the turbulence characteristics being spatially uniform. The stagnating turbulence is certainly non-uniform [13, 14]. However, to quantitatively predict the behavior of turbulence characteristics near the stagnation point, an adjustment of the classical turbulence model constants is required even in constant density cases [14]. Modeling the effects of heat release on a stagnating turbulence further complicates the problem and involves additional constants [13]. One could try to combine Eq. 1 with a sophisticated turbulence submodel, however, such a combined model would be quite complex and would include a large set of adjustable constants. The presence of adjustable constants questions any conclusions drawn by comparing numerical and measured data. However, such complications can be avoided if the focus is placed on our primary goal: To test whether or not Eq. 1, which is applicable to developing flames only, can predict the basic features of stagnation turbulent flames, in particular, the stationarity of such flames and the spatial uniformity of δ_t. We assume that the above simplification is adequate for this goal.

In addition to this simplification, the following standard assumptions [13]

$$\tilde{u}(x,r,t') = w_1 U(z,t'); \quad \tilde{v}(x,r,t') = w_1 \frac{r}{d} V(z,t'); \quad \bar{p}(x,r,t') = \rho_u w_1^2 \left[P(z,t') - \frac{1}{2}\left(\frac{r}{d}\right)^2 Q \right];$$

$$\tilde{c}(x,r,t') = C(z,t'); \qquad R \equiv \frac{\bar{\rho}}{\rho_u} = \frac{1}{1+\gamma\tilde{c}} \quad (2)$$

have been invoked. Here, $t' = t/\tau_t$ is dimensionless time, u and v are axial and radial mean

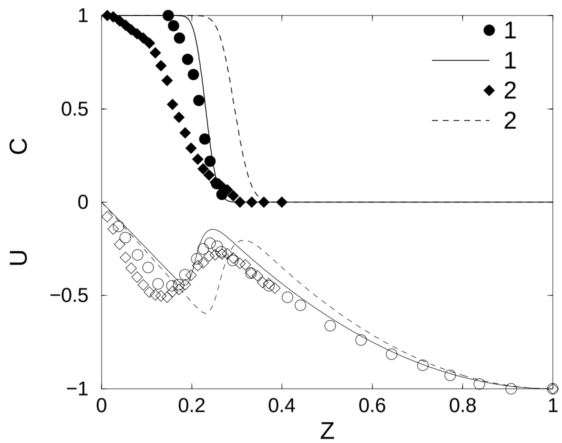

Figure 1: Dimensionless axial velocity U and progress variable C vs. dimensionless distance z. Symbols show the experimental data: 1 - Ref. [15]; 2 - Ref. [16]. Curves have been computed. The computed U-profiles are Favre-averaged, other results are Reynolds-averaged.

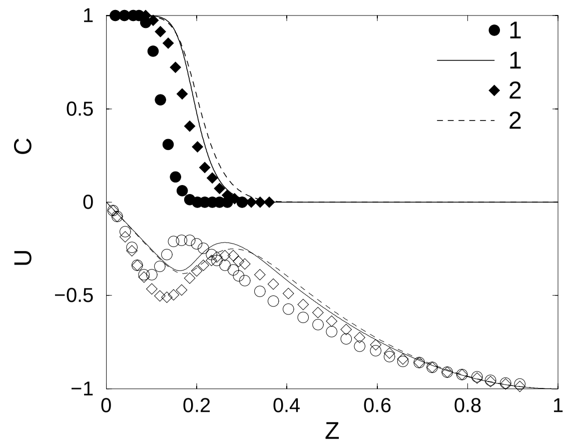

Figure 2: Dimensionless axial velocity U and progress variable C vs. dimensionless distance z. Symbols show the data [17] measured with two grids with different hole diameters, h: 1 - $h = 4$ mm; 2 - $h = 6$ mm. Curves have been computed. All the results are Favre-averaged.

velocities, respectively; r is radial distance; $z = x/d$ is dimensionless axial distance ($z = 0$ at the stagnation point); d is the distance between the jet exit and the stagnation point; w_1 is the mean axial velocity at the jet exit; p is the pressure; and $\gamma = \rho_u/\rho_b - 1$.

Then, the studied, axially symmetrical flames were modeled by the following equations:

$$\begin{aligned} \Pi\frac{\partial R}{\partial t'} + 2RV + \frac{\partial RU}{\partial z} &= 0; \\ \Pi R\frac{\partial V}{\partial t'} + RV^2 + RU\frac{\partial V}{\partial z} &= Q + \frac{D_t}{w_1 d}\frac{\partial}{\partial z}\left(R\frac{\partial V}{\partial z}\right); \\ \Pi R\frac{\partial C}{\partial t'} + RU\frac{\partial C}{\partial z} &= \frac{D_t}{w_1 d}\frac{\partial}{\partial z}\left(R\frac{\partial C}{\partial z}\right) - A\frac{u_1'}{w_1}Da^{1/4}\frac{\partial C}{\partial z}; \end{aligned} \tag{3}$$

supplemented with the following boundary conditions:

$$U(1) = -1; \quad U(0) = V(1) = \frac{\partial V}{\partial z}(0) = C(1) = \frac{\partial C}{\partial z}(0) = 0. \tag{4}$$

Here, $\Pi = d/(w_1\tau_t)$. The parameter Q has to be calculated as part of the numerical solution in order to satisfy the extra boundary condition $U(0) = 0$. The turbulence characteristics at the jet exit, u_1' and L_1, reported in the experimental papers, have been used in order to quantify $D_t = 0.09k_1^2/\epsilon_1$ with $k_1 = 1.5u_1'^2$ and $\epsilon_1 = 0.37u_1'^3/L_1$. The only constant A of the model is equal to 0.5. The same value of this constant has been used by different authors [2, 4, 6, 7, 8, 9] when validating the FSC model under different conditions.

Results and discussion

Figures 1-3 summarize the results which are representative of many computational tests performed by us but not shown here due to space limitations. The differences between the measured and computed results are not surprising, bearing in mind the simplifications of the turbulence submodel and the following two limitations of Eq. 1. First, since Eq. 1 predicts zero burning velocity at the limit of $u' \to 0$, the domain of validity of this Equation is associated with $u'/S_L \geq 1$ [2, 3]; whereas the experimental data correspond to $u'/S_L \leq 1$. Second, a flamelet

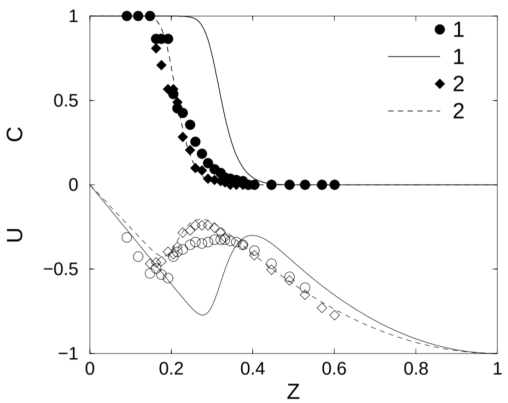

Figure 3: Dimensionless Favre-averaged axial velocity U and progress variable C vs. dimensionless distance z. Symbols show the experimental data [18]: 1 - $w_1 = 3$ m/s; $u_1' = 0.33$ m/s; $S_L = 0.4$ m/s; 2 - $w_1 = 2.25$ m/s; $u_1' = 0.18$ m/s; $S_L = 0.3$ m/s. Curves have been computed.

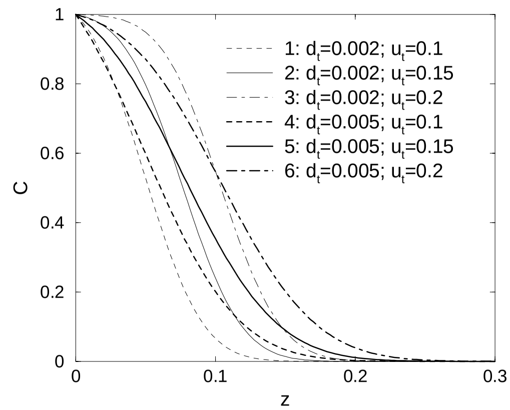

Figure 4: Dimensionless progress variable C vs. dimensionless distance z. Curves have been computed from Eq. 6 with constant B being adjusted in order to get $C(0) = 1$. The dimensionless parameters of the calculations, u_t and d_t, are presented in the legends.

quenching submodel is not included in Eq. 1[1] and the difference between the measured and computed results in case 2 in Fig. 1 is, in part, associated with this limitation.

Despite the above simplifications, certain properties of stagnation turbulent flames are well predicted, e.g., the profiles 1 in Fig. 1, 2 in Fig. 2, and 2 in Fig. 3. In other cases (e.g., 1 in Fig. 2 and 1 in Fig. 3), the computed progress variable profiles are more distant from the stagnation point than the measured ones, however, the value of the local minimum of $|U|$, which is often associated with turbulent flame speed, is predicted much better in these cases.

For our purposes, the following results are of the most importance. First, the profiles shown in Figs. 1-3 have been computed by solving the unsteady Eqs. 3, and all these profiles are asymptotically steady at $t \to \infty$. Thus, the FSC model is able to predict a steady planar[2] flame in a stagnation point mean flow. Second, in all the reported cases, with the exception of case 2 in Fig. 1, the mean flame brush thickness, δ_t, is predicted quite well despite the fact that Eq. 1 yields a permanently growing δ_t in the classical planar one-dimensional case.

These results imply that, due to the spatial non-uniformity of the unburned mixture flow, properties of premixed turbulent stagnation flames may differ substantially from the properties of classical planar one-dimensional fully developed flames which propagate through an initially uniform mixture. To further illustrate these claims, let us consider Eqs. 3 in the simplest steady and constant density case. Then, Eqs. 3 are reduced to

$$U = -2z\left(1 - \frac{z}{2}\right); \qquad V = 1 - z; \qquad Q = 1; \tag{5}$$

$$\frac{dC}{dz} = -B \exp\left(\frac{u_t z - z^2 + z^3/3}{d_t}\right), \tag{6}$$

where $u_t = A u_1' Da^{1/4}/w_1$ and $d_t = D_t/(w_1 d)$ are the parameters of the problem and B is a constant. Although Eq. 6 cannot satisfy the boundary conditions of $dC/dz(0) = dC/dz(1) = 0$ exactly; slightly modified conditions of $dC/dz(0) = \varepsilon$ and $dC/dz(1) = \varepsilon_1$, where $0 < \varepsilon \ll 1$ is an arbitrary small parameter and $\varepsilon_1 = \varepsilon \exp\left((u_t - 2/3)/d_t\right) \ll \varepsilon$, can be satisfied, if d_t is small enough. For instance, Fig. 4 shows numerical solutions of Eq. 6 at various d_t and u_t, the

[1] Such a modification of the FSC model is discussed elsewhere [2].

[2] According to Eq. 2, $\partial\tilde{c}/\partial r = 0$ and, thus, the flame structure is locally planar.

values of these parameters corresponding to typical conditions in stagnation flames. A profile calculated in case 3 is typical for such flames. In cases 2 and 6, the profiles are similar to the data of Cheng et al. [16] (see diamonds in Fig. 1). This similarity implies that quenching of flamelets by turbulent eddies is not the only possible cause of flame drift to the stagnation point. This issue requires further study. Finally, the maximum slopes of the calculated curves are controlled mainly by d_t (cf. fine and bold curves) but they are weakly affected by u_t (cf. curves 1, 2, and 3). Since d_t is controlled not only by turbulence characteristics but also by the distance d, this property is quite specific for the solutions discussed as compared with the solutions of the classical planar one-dimensional balance equations.

Conclusions

Numerical simulations of premixed turbulent stagnation flames have been performed using the FSC model. The results indicate that the model well predicts the stationarity of these flames and their mean thickness. This suggests that premixed turbulent stagnation flames are not equivalent to classical planar one-dimensional *fully developed flames* which propagate through an initially uniform mixture.

Acknowledgment

This work was supported by the Chalmers Combustion Engine Research Center and by the Swedish Research Council for Engineering Sciences (grant 224-99-371).

References

[1] V. L. Zimont. *Exp. Therm. Fluid Sci.*, 21:179–186, 2000.

[2] V. Karpov, A. Lipatnikov, and V. Zimont. *Proc. Combust. Inst.*, 26:249–257, 1996.

[3] A. Lipatnikov and J. Chomiak. *J. Engines, SAE Transactions*, 106:2441–2453, 1997.

[4] L. Maciocco and V. Zimont. *20-th Annual Meeting of the Italian Section of the Combustion Institute "Frantic97"*, pp.X-2.1–2.4, 1997.

[5] A. Lipatnikov, J. Wallesten, and J. Nisbet. *COMODIA 98,* JSME, pp.239–244, 1998.

[6] P. Birkestad. Modeling of Premixed Turbulent Combustion for Afterburners and Low Emission Combustors. M.S. Thesis. Chalmers University of Technology, 1998.

[7] F. Dinkelacker and S. Hölzler. *Combust. Sci. Techn.*, 158:321-340, 2000.

[8] V. Zimont, F. Biagioli, and K. Syed . *Int. J. Progress Comp. Fluid Dyn.*, 1:14, 2000.

[9] V. Zimont, W. Polifke, M. Bettelini, and W. Weisenstein. *J. Engng. Gas Turbines Power*, 120:526–532, 1998.

[10] N. Peters. *Turbulent Combustion.* Cambridge University Press, Cambridge, 2000.

[11] A. Lipatnikov and J. Chomiak. *Combust. Sci. Techn.*, 2001, in press.

[12] A. Lipatnikov and J. Chomiak. *Proceedings of the Combustion Institute* 28: 2000.

[13] K. N. C. Bray, M. Champion, P. A. Libby. *Comb. Flame*, 120:1–18, 2000.

[14] M. Champion and P. A. Libby. *Phys. Fluids*, 6:1805–1819, 1994.

[15] P. Cho, C.K. Law, R.K. Cheng, and I.G. Shepherd. *Proc. Combust. Inst.*, 22:739-745,1988.

[16] R. K. Cheng, I. G. Shepherd, and Gökalp I. *Comb. Flame*, 78:205–221, 1989.

[17] S. C. Li, P. A. Libby, and F. A. Williams. *Proc. Combust. Inst.*, 25:1207–1214, 1994.

[18] E. J. Stevens, K. N. C. Bray, and B. Lecordier. *Proc. Combust. Inst.*, 27:949–955, 1998.