[2] Edelman R. B. *Proc. of NSF (RANN) Workshop on Numerical Simulation of Combustion for Appl. to Spark and Compres. Ign. Engines,* SAI, La Jolla, Calif., 1975.

[3] Reinolds W. C. *STANJAN Interactive Computer Programs for Chemical Equilibrium Analysis,* Stanford Univ., 1986.

3D MODELING OF PULSED JET COMBUSTION

S. M. Frolov*, M. Suffa†, R. Tatschl‡, P. Wolanski§

* *Semenov Institute of Chemical Physics, Moscow, Russia*
† *Hochschule fur Wirtschaft und Technik, Dresden, Germany*
‡ *AVL LIST GmbH, Graz, Austria*
§ *Politechnika Warszawska, Warszawa, Poland*

Introduction

The concept of Pulsed Jet Combustion (PJC), proposed by Oppenheim [1] as an alternative mode of energy release in stratified-charge internal combustion engines, shows a number of practical advantages. Based on the fundamental ideas of Gussak, Semenov and Zel'dovich concerning the role of active radicals in the torch mechanism of ignition and flame propagation, the PJC concept implies a staged combustion process:

- the flame generated in a prechamber is extinguished by shear at efflux from the orifice between the prechamber and the main chamber;
- combustion is spontaneously reinitiated in the main chamber in the core of the hot turbulent jet.

According to [1], further chemical transformations in the main chamber occur in a fireball, i.e. via distributed combustion sustained by multiple pockets of high concentration of active radicals. Of prime importance are the entrainment processes resulting in the fireball growth.

It should be emphasized that the fireball mode of combustion differs in principle from turbulent propagating flame mode. The latter is characterized by laminar-like average structure with certain characteristic thickness of the turbulent reaction zone. The turbulent propagating flame mode is assumed in the most stratified charge concepts.

This paper reports a 3D computer modeling of the PJC system behavior, focusing on the evolution of exothermic processes.

PJC System Geometry

The PJC system is assumed to consist of the main combustion chamber and two identical opposed jet prechambers. The main combustion chamber is a cylinder of 8.26 cm bore, 10 cm in height, and total volume ≈ 536 cm^3. The prechamber is of the same design as reported in [1]. It consists of a cylindrical cavity ≈ 1 cm^3 in volume and a co-axial tubular electrode perforated (32 orifices) for charge inlet. The orifice plates between the prechambers and the main combustion chamber are 2.5 mm in diameter. The orifices are sharp-edged, 0.5 mm thick. The prechambers are affixed to the main chamber at height 9 cm. Initially, the PJC system is filled with propane–air mixture of equivalence ratio 0.6 at pressure 5 bar and temperature 60 °C.

At $t > 0$, propane-air mixture of equivalence ratio 1.5 is injected through the perforations in the prechambers. The injection persists for 10 ms at the mass flow rate 0.332 kg/s. At $t = t_{ign} = 10$ ms, the rich mixture remaining in the prechambers is ignited by a spark. The ignition point is located at the outlet orifice inside the prechamber.

Modeling

A set of 3D conservation equations of mass, momentum and thermal energy serves as the basis of the mathematical model.

The variables in the governing equations are expressed in terms of mean and fluctuating components. The subsequent averaging process generates unknown correlations of the products of the fluctuating components. The unknown correlations are modeled within the framework of the standard k–ε model. The k–ε model involves additional partial differential equations for the turbulent kinetic energy k and its dissipation rate $\varepsilon = C_\mu^{3/4} k^{3/2}/l_t$, where $C_\mu = 0.09$ is an empirical constant and l_t is the turbulence length scale. The initial conditions for the turbulence model in the present case are: $l_t = 1$ mm, $k = 1$ m^2/s^2. Appropriate boundary conditions are also imposed. Semi-empirical "laws of the wall" are invoked for the dependent variables in the near wall region.

Turbulence-Controlled Combustion Model

The combustion model employed is the turbulence-controlled model proposed in [2]. This model assumes that, in premixed turbulent flames, the reactants (fuel and oxygen) are contained in the same eddies and are separated from the eddies containing hot combustion products. Since the chemical time scales are very short, as compared with those of turbulent transport processes, it can be assumed that the rate of combustion is determined by molecular mixing of the eddies containing reactants with those containing hot products, in other words, by the rate of dissipation of these eddies.

The mean reaction rate can thus be written, in accordance with [2], as

$$\overline{\rho \dot{r}_{fu}} = \frac{C_{fu}\overline{\rho}}{\tau_R} \min\left(\overline{y}_{fu}, \frac{\overline{y}_{Ox}}{S}, \frac{C_{Pr}\overline{y}_{Pr}}{1+S}\right)$$

The first two terms of the 'minimum value of' operator $\min(\overline{y}_{fu}, \overline{y}_{Ox}/S, (C_{Pr}\overline{y}_{Pr})/(1+S))$ simply determine whether fuel or oxygen is the limiting reactant, and the third is a reaction probability which ensures that the flame does not exist in the absence of hot products. C_{fu} and C_{Pr} are empirical coefficients and τ_R is the turbulent mixing time-scale. The calculations were made for the following values: $C_{fu} = 20$, $C_{Pr} = 0.5$, $\tau_R = 5 \cdot 10^{-3}$ s.

The ignition event was modeled by introducing a kernel of hot combustion products of fixed size at a certain location.

Numerical Solution

The FIRE code developed by AVL was used to solve the governing equations. FIRE rewrites all the equations in a general curvilinear non-orthogonal coordinate system. The individual terms of partial differential equations are replaced by algebraic expressions obtained by integration over a control volume of finite size. The finite volume discretization leads to a set of nonlinear algebraic equations for the values of dependent variables, such as velocities, pressure etc., at the centre of the computational mesh.

Results and Discussion

The computational grid contained about 16,000 spatial cells. The predicted density distributions were compared directly to the schlieren records of jet plumes obtained by means of the dual PJC system of [1], similar to the one used in our study. A comparison between the predicted and measured data have shown a good quantitative agreement until $t = 15$ ms, which is an indication of satisfactory simulation.

Figure 1 shows the predicted spatial profile of the reaction rate at $t = 12.5$ ms. It is evident that the exothermic process in the main combustion chamber occurs over the entire core of the turbulent jet (i.e., in the fireball mode). In contrast to the combustion process in the main chamber, the process in the prechamber exhibits features of a turbulent flame. The maximum outflow velocity from the prechamber is about 57 m/s at $t = 12.5$ ms.

Figure 2 shows the spatial profile of the reaction rate at $t = 15$ ms. Apparently, the distributed exothermic process typical for $t = 12.5$ ms is replaced now by a distinctly propagating mode of combustion. Flame extinction is apparent at the exit of the high-speed jet. In the prechamber, the flame propagates counter to the mass flow. Therefore, the apparent flame velocity is substantially lower than in the main chamber, despite the fuel-rich mixture composition in the former. Note that the thickness of the turbulent flame in the main chamber is affected by the computational mesh size which grows towards the periphery of the chamber.

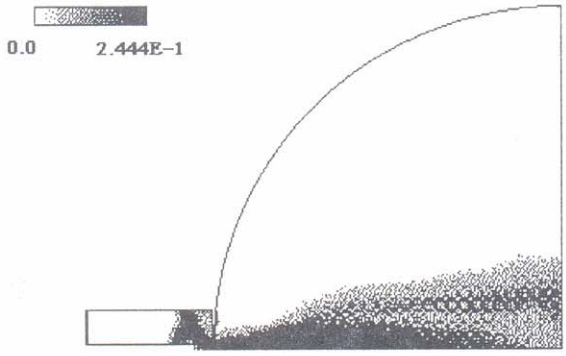

Figure 1: Predicted reaction rate distribution in a single-orifice dual PJC system at 2.5 ms after ignition.

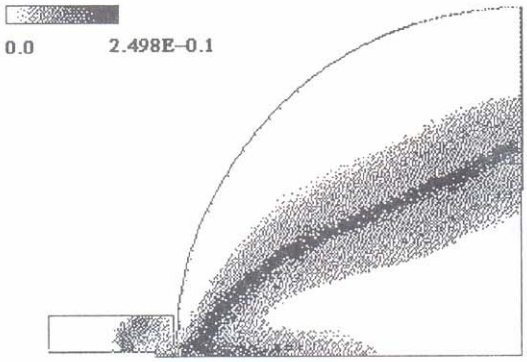

Figure 2: Predicted reaction rate distribution in a single-orifice dual PJC system at 5 ms after ignition.

Conclusion

As a result of 3D numerical simulation of the PJC system, it has been found that turbulent combustion in the main chamber exhibits two distinct modes, namely the fireball mode and the turbulent diffusion flame mode. The fireball mode is characteristic of the initial stage of the process, while the turbulent flame dominates at later stages. Further studies are planned with the view of improving the combustion model.

References

[1] D. M. Hensinger, J. A. Maxson, K. Hom, A. K. Oppenheim. *SAE Paper 920414*, 1992.

[2] B. F. Magnussen, B. H. Hjertager. *Proc. 16th Symp. (Int.) on Comb.*, Pittsburgh, Pennsylvania, 1977.

TURBULENT DIFFUSION FLAME WITH A MEAN STREAMWISE PRESSURE GRADIENT

V. Frost*, M. Gorokhovski†, Yu. Vladimirov*

**Institute for Problems in Mechanics, Russian Academy of Science, Russia*
† Universite' de Rouen — U.R.A. C.N.R.S. 230 CORIA, France

I. The strong nonlinearity of chemical reaction rates as the main problem of turbulent combustion theory was first emphasized by Ya. Zel'dovich (1949): the mean reaction rate cannot be correctly evaluated in terms of mean flow properties. After the advent of probability density function (PDF) methods, the problem of reaction rate averaging in turbulent combustion modelling appears to be simplified. Over the last 30 years, the transport equations for the PDF of temperature and concentrations have been obtained [1–3] with the view of turbulent combustion applications.

Combustion–flow interactions related to the flame present a similar problem: due to the strongly nonmonotonic form of density function, an additional density — velocity correlation must be calculated. The latter arises as a direct effect of mean streamwise pressure gradient on the turbulent variable-density flow through selective acceleration of low and high density fluids. This problem, known as the one of flame-generated