THREE-DIMENSIONAL MODELLING OF MIXTURE FORMATION IN DIRECT INJECTION DIESEL ENGINES

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ABSTRACT

Results of three-dimensional computations of direct injection of diesel fuel in a multi-hole injector in diesel engine are presented. A 6-cylinder (DDC 6V-TA), direct injection (DI) diesel engine is simulated where the study is focused the fuel/air distribution prior to ignition. The cylinder bore and stroke dimensions are 123mm and 127mm respectively. The engine is rated 224kW at 1200rpm. The influence of fuel jet and air swirl level on the fuel/air mixing was investigated, for the engine speed of 1200rpm at 60kW load. From the results it was observed that turbulence generated by the fuel spray jet was significantly responsible for the fuel/air mixing particularly for the fist one millisecond of the fuel injection. However swirl influences the formation of fuel/air mixture and it was found that the optimal swirl ratio was found to lie between 0 and 5.

INTRODUCTION AND BACKGROUND

An internal combustion engine cylinder is a complex system with a number of physical and chemical processes occurring simultaneously. These processes include sub-sonic flow of the fluid, unsteady turbulent flow of the mass, momentum and energy exchange. Others include wall heat transfer, turbulent flame propagation, wall quenching, pollutant formation, and for diesel engine, the spray dynamics. The overall performance and emissions of the engine is a complex interaction among these efects. Fuel/air mixing of a spray is the combined effect of fuel evaporation and turbulent mixing in the combustion chamber. Faster evaporation of the liquid fuel does not necessarily increase mixing because it can hinder spray penetration and hence retarding mixing rate.

In compression ignition (CI) engine, fuel is injected by fuel injector into the engine combustion chamber towards the end of compression stroke just before the desired start of combustion. The liquid fuel is injected at high pressure jets where it is atomized. The atomized fuel evaporates and mixes with air as it penetrates in the combustion chamber Since the chamber temperature (and pressure) is high, the fuel/air mixture ignites spontaneously after a few degrees crank angle of ignition delay Ignition delay is defined as the time elapsed from the dynamic start of injection to the noticeable pressure rise.

The flow pattern within a cylinder of internal combustion (IC) engine is the most important factor controlling the combustion process. The fuel/air mixing, burning rate and flame propagation are strongly influenced by the in-cylinder flow field. The common in-cylinder flows are: (i) the large scale rotating flows (swirl) established by the conical intake jet; (ii) the swirl produced and enhanced by compression and combustion processes, and (iii) the squish motion produced by the squish chamber Swirl is defined as the tangential air motion about the cylinder axis. Squish is the radial air motion perpendicular to the cylinder axis and it enhances turbulence close to top dead center (TDC). Swirl level is necessarily be optimal for flame kernel growth and flame propagation. High swirl levels are undesired because would cause the whole charge to rotate (solid body) with the fuel-rich mixture areas inhibiting further fuel/air mixing. It has been reported by Ho and Santavicca [1987] that increased turbulence for lean mixtures results to negative flame stretch thus decelerating combustion. Flame stretch is caused by the velocity gradient of the turbulent flame.

Fuel injection characteristics (e.g. injection angle, duration, etc.) have often been described as the vital factor in performance of diesel engine. The fuel injection characteristics need to be matched with the air motion and the operation conditions of the engine. The matching aims at having a "right" fuel/air mixture correctly distributed in the combustion chamber at the desired time. This is to say the spatial and temporal distribution of combustible mixture is of vital importance for the engine performance and emissions. Thus a degree of control is very important to operated direct injection (DI) engine than merely to obtain an overall equivalence ratio. Equivalence ratio is defined as the ratio between the actual fuel/air and the stoichiometric fuel/air ratios.

MIXTURE FORMATION IN DIESEL ENGINES

Computer modelling of mixture formation and combustion processes in direct injection (DI) engines has advanced lately, such that in-cylinder processes can be predicted within acceptable accuracy. Several phenomelogical models to predict spray behaviour and turbulent mixing for diesel combustion engines have been proposed by Mansouri [1982] and Kono [1985]. Nishida [1989] pointed out that higher swirl levels results into very lean mixtures which are not combustible. The fuel/air mixing due to jet generated turbulence has been shown (Mtui [1995]) to strongly dependent on the injection angle (relative to the cylinder head) and velocity Detailed multi-dimensional modelling has been introduced (Gosman [1982] and Amseden [1989]) which accounts for the in-cylinder fluid dynamics and fuel spray interactions.

However, publications such as by Monagan [1981] andYoshikawa [1989] reveal that the DI diesel engine poses a major challenge to design engineers due to the need to correctly "shape" and match the air motion and fuel injection. Of particular importance to engine performance and emissions is the distribution of vapour fuel between the lean, flammable and rich mixtures in the combustion chamber. The fuel distribution in the chamber has been found useful in direct injection of diesel in engines by isolating controlling physical processes (Abraham and Bracco [1992]). It is precisely the potential of multi-dimensional modelling to provide detailed spatial and temporal information of velocity, temperature, concentration of fuel, etc. Computed results allow comparatively speedy and inexpensive exploration of design changes.

A major challenge in diesel engine combustion is the design to achieve rapid fuel/air mixing so as to complete combustion in a desired combustion duration for maximum thermal efficiency. In view of this challenge the objectives of this paper are to establish the range of swirl ratios for optimal enhancement of fuel/air mixing, and to determine the enhancement of fuel/air mixing through the turbulence generated by the jet shear. Swirl ratio is defined as the ratio of the tangential inlet air velocity to the engine rotational speed. Swirl is largely responsible for the turbulence generation in IC engines.

THE GOVERNING EQUATIONS

The governing equations are written in vector form for compactness. The gas phase model solves for the species concentrations, momentum and energy. The time-averaged quantities are given by equations 1 through 5. The conservation of mass for species is given by

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \bar{u}) = \rho^{s} \tag{1}$$

where ρ is the specie density and ρ^s is the source term due to fuel spray. The momentum equation is given by

$$\frac{\partial}{\partial t}(\rho \bar{u}) + \nabla(\rho \bar{u} \bar{u}) = \nabla p + \nabla(\Gamma_{\mu} \nabla \bar{u}) + F^{s} + \rho g \tag{2}$$

where p is the fluid pressure, F^s is the momentum gained per unit volume due to spray and g is the gravitational acceleration. Γ_{μ} is the effective momentum diffusivity given by

$\Gamma_{\mu} = \mu_l + \mu_t$ = laminar viscosity + turbulence viscosity
$\mu_t = C_{\mu} \rho\ k^2 \varepsilon$ with
k = turbulence kinetic energy
ε = rate of dissipation of kinetic energy

The conservation of energy of the compressible flow is given by

$$\frac{\partial}{\partial t}(\rho I) + \nabla(\rho \bar{u} I) = \nabla(\Gamma_{I} \nabla T) + \dot{Q}^{s} + \dot{Q}^{c} \tag{3}$$

where I is the specific enthalpy (excluding chemical energy due to combustion) and Q^c and Q^s are the rate of energy release due to combustion and the rate of energy exchange of the gas with spray droplets respectively. The local temperature is denoted by T and GI is the effective diffusivity of energy and is given by

$\Gamma_l = (\mu/\sigma)_l + (\mu/\sigma)_t$ where σ is the Prandtl number.

The turbulence is modelled by the standard k-ε where k is the turbulence kinetic energy (TKE) and e is the rate of dissipation of TKE. The equations for the conservation of k and e are shown in equations 4 and 5 respectively (4)

$$\frac{\partial}{\partial t} + \nabla(\rho\bar{u}k) = \nabla(\Gamma_k \nabla k) + S_k \tag{5}$$

$$\frac{\partial}{\partial t} + \nabla(\rho\bar{u}k) = \nabla(\Gamma_\varepsilon \nabla \varepsilon) + S_\varepsilon \tag{6}$$

where

$S_k = \mu_t G_k - \rho\varepsilon$

$$S_\varepsilon = C_1 \frac{\varepsilon}{k}\mu_t G_k - C_2 \frac{\rho}{k}\rho\varepsilon + C_3 \frac{G^{2k}}{\varepsilon} \tag{7}$$

and the generation of turbulence due to velocity gradient is given by

$$G_k = (\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})\frac{\partial u_i}{\partial x_j} \tag{8}$$

Γ_k and Γ_ε are the effective diffusivities of k and e respectively

The Numerical Solution

The governing equations are solved for the finite difference approximation discritized both in space and time (Amseden [1987]). The spatial discritization of the computation domain is constructed using a finite volume method which largely preserves the local conservation properties of the differential equations. The cells are arbitrary hexahedron which allow the description of complex boundaries like doomed cylinder head and piston bowls. The solution of the gas phase is Lagrangian-Eularian. The computational time step is selected in every cycle based on accuracy criteria. The piston bowl computational meshes are body-fitted while those above the piston bowl moves with the piston. During the compression and expansion stokes the meshes are respectively chopped of and restored to improve computational efficiency. Figure 1 below depicts the computational grid arrangement for a cross-section area of the combustion chamber close to TDC. The injector is located at the shaded area. Diesel fuel is injection at 2 degrees crank angle before top dead center (BTDC) at a velocity of 150m/s.

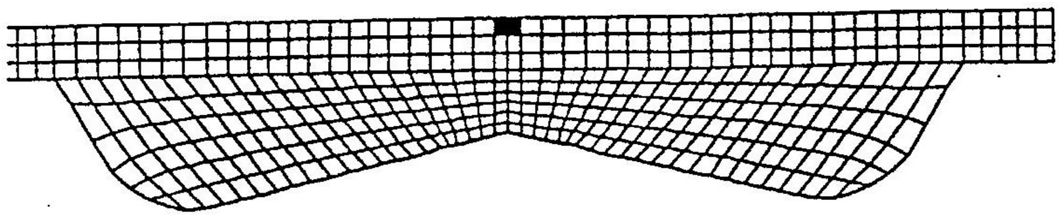

Fig. 1 Computational Grid Arrangement

Initial Conditions

Prior to the start of computation, initial and boundary conditions are establish to match engine conditions.

At the start of compression stroke boundary conditions were established. Initial velocity distribution is provided by the piston speed with engine running at 1200rpm. The air inlet temperature and swirl are defined. The initial pressure is explicitly expressed by species temperature and density in the equation of state. Initial pressure, temperature and turbulence kinetic energy are assumed to be uniform in the computational grids. Turbulence kinetic energy, kIVC and its rate of dissipation eIVC (at the intake valve closure) were initialized using the equations of Grasso et al [1987] as shown below

$$k_{IVO}=0.1*U^2 \quad \text{and} \quad \varepsilon_{IVO} = 0.05*U/A^{0.5}$$

where U and A are the average intake velocity and valve open are respectively.

Boundary Conditions

The temperature of the air at the wall are set equal to those of the wall temperature TW which is kept constant. The wall velocity and the pressure gradient normal to the wall are respectively set to zero. The law-of-the-wall functions which were originally proposed by Launder [1974] were assumed. These wall functions are used to establish the fluxes of momentum and energy at the wall. This is accomplished by matching the computed fluid velocities and temperature at the grid points adjacent to the wall. The wall shear stress (τ_w) is calculated from the frictional velocity

($\tau_w=\rho^{*2}$). Laminar velocity profile is invoked in the laminar sub-layer; it is expressed as.

$$\frac{U}{U^*}=\sqrt{(\frac{yU}{\nu})} \tag{9}$$

In fully turbulent region, the logarithmic law velocity is used as expressed in equation 10.

$$\frac{U}{U^*}=\frac{1}{\kappa}\ln(\frac{yU^*}{\nu})+B \tag{10}$$

where U* is the frictional velocity and k is the von Karman constant ($\approx$0.4327).

The wall heat flux J_w is determined by Reynolds analogy as:

$$\frac{J_w}{\rho U^* C_p(T-T_w)}=\frac{1}{P_{rL}(\frac{U}{U^*})} \tag{11}$$

where u* is the frictional velocity, T is the local gas temperature, P_{rL} is the laminar Prandtl number and C_p is the specific heat at constant temperature.

The turbulence kinetic energy, k and the rate of dissipation of kinetic energy, ε are defined below as

$$k=C_{\mu\varepsilon}(\frac{\tau_w}{\rho_w}) \tag{12}$$

$$\varepsilon=C_{\mu\varepsilon}(\frac{k^{3/2}}{y}) \tag{13}$$

where k and ε are evaluated at a distance y from the wall and $C_{\mu\varepsilon}$ is an empirical constant.

RESULTS AND DISCUSSIONS

The results discussed below are categorized in swirl ratio levels, i.e. 0, 2.5 and 5.0. The in-cylinder flow (Figs 2 to 4) are for case studies at swirl ratio of 2.5. The contour results in Figs 2 to 4 are labelled by H (High or

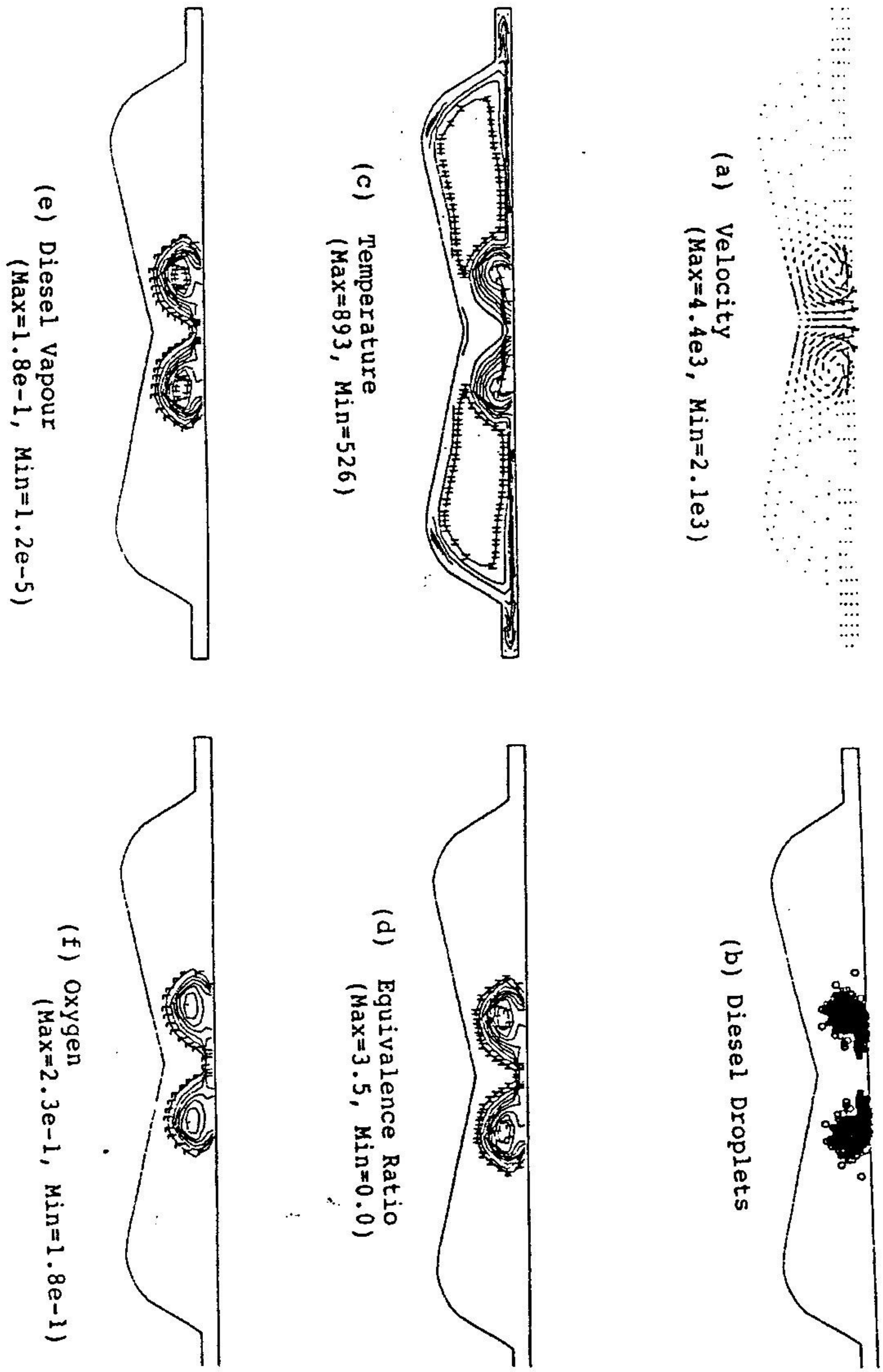

Fig. 2: In cylinder flow processes (5 deg After TDC)

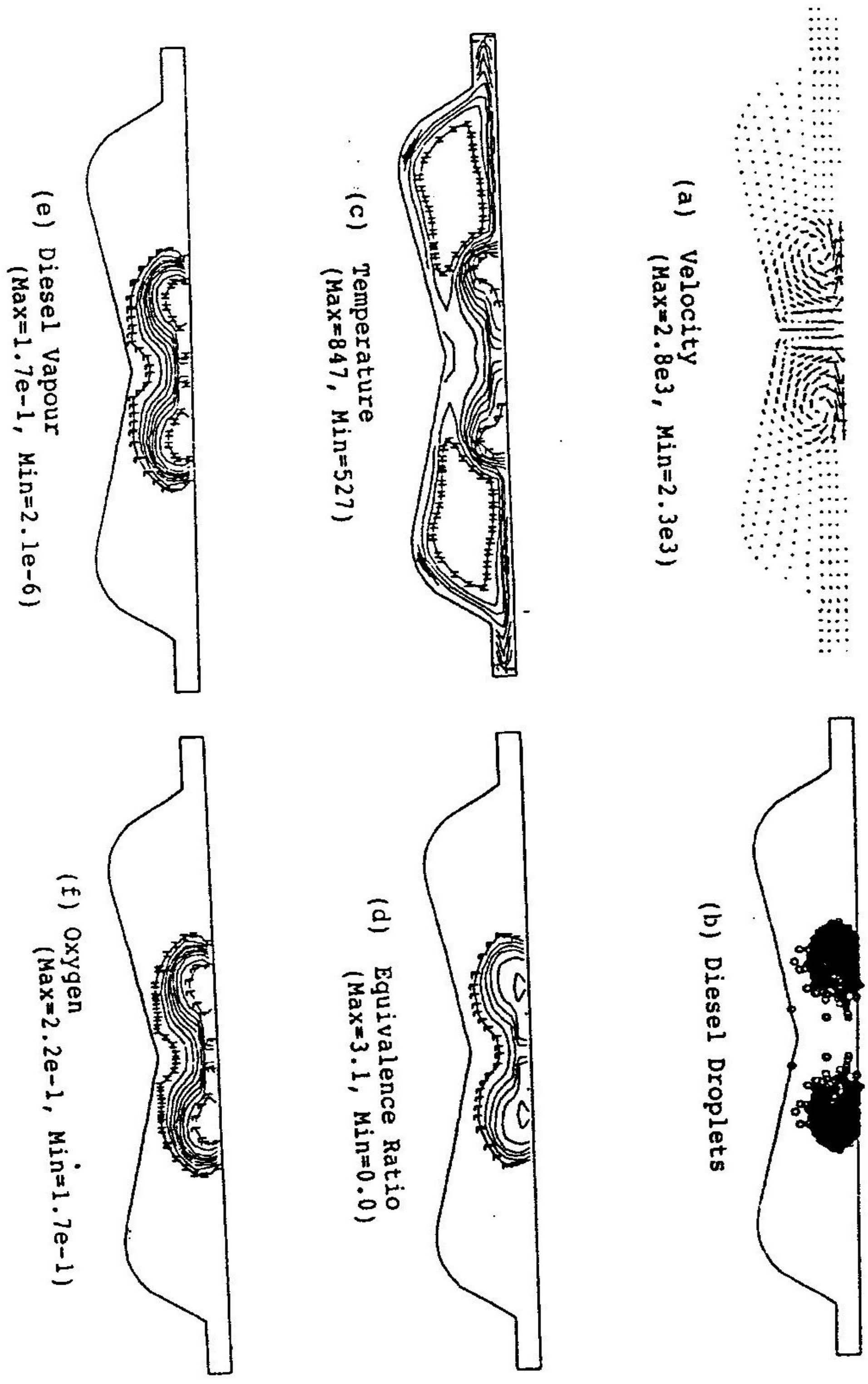

Fig. 3: In-cylinder Flow processes (10 deg after TDC)

Maximum) and L (Low or Minimum). The fuel vapour in the chamber are categorized in the level of equivalence ratios (f). For lean mixture $0 \leq \phi \leq 0.5$ and for flammable mixture $0.5 \leq \phi \leq 2.0$. The rich mixture is defined for $\phi \geq 2.0$.

Velocity Distribution

The resultant velocity of air in the chamber are shown in units of cm/sec. Squish velocity is very prominent especially close to the chamber center (Figs. 2a, 3a and 4a). Velocity diminishes as the piston moves from TDC to bottom dead center (BDC) because of turbulence dissipation, and thus no other source of turbulence generation. This velocities are responsible for fuel/air mixing.

Diesel Droplets

Diesel droplets are shown in Figs 2b - 4b. Close to TDC (Fig 2b) the droplets are dense and close to the chamber center At 20 degrees later the droplets have already transported downstream, dispersed and evaporated. The interaction of swirl and squish motions together with the spray velocity transport the fuel drops in the chamber

Temperature Distribution

Temperature distribution in the chamber are shown in Figs 2b, 3b and 4c. The temperature (in Kelvin) in the chamber is observed to drop as the piston descends to BDC. It is interesting to note that temperature is significantly low in the vicinity of the fuel jet due to fuel evaporation. However, due to thermal diffusion, the temperature in the zones of flammable mixture are within the ignition temperature of diesel fuel (about 900K).

Equivalence Ratio

The equivalence ratio is shown in Figs 2d to 4d decreasing with increasing residence time. As the piston descends to BDC the time elapsed allows the formation of combustible mixture. The decreasing fmax from 3.5 (Fig 2d) to 3.0 (Fig 4d) indicates mixing has taken place to form combustible mixture from the initially rich mixture (non combustible). It is observed

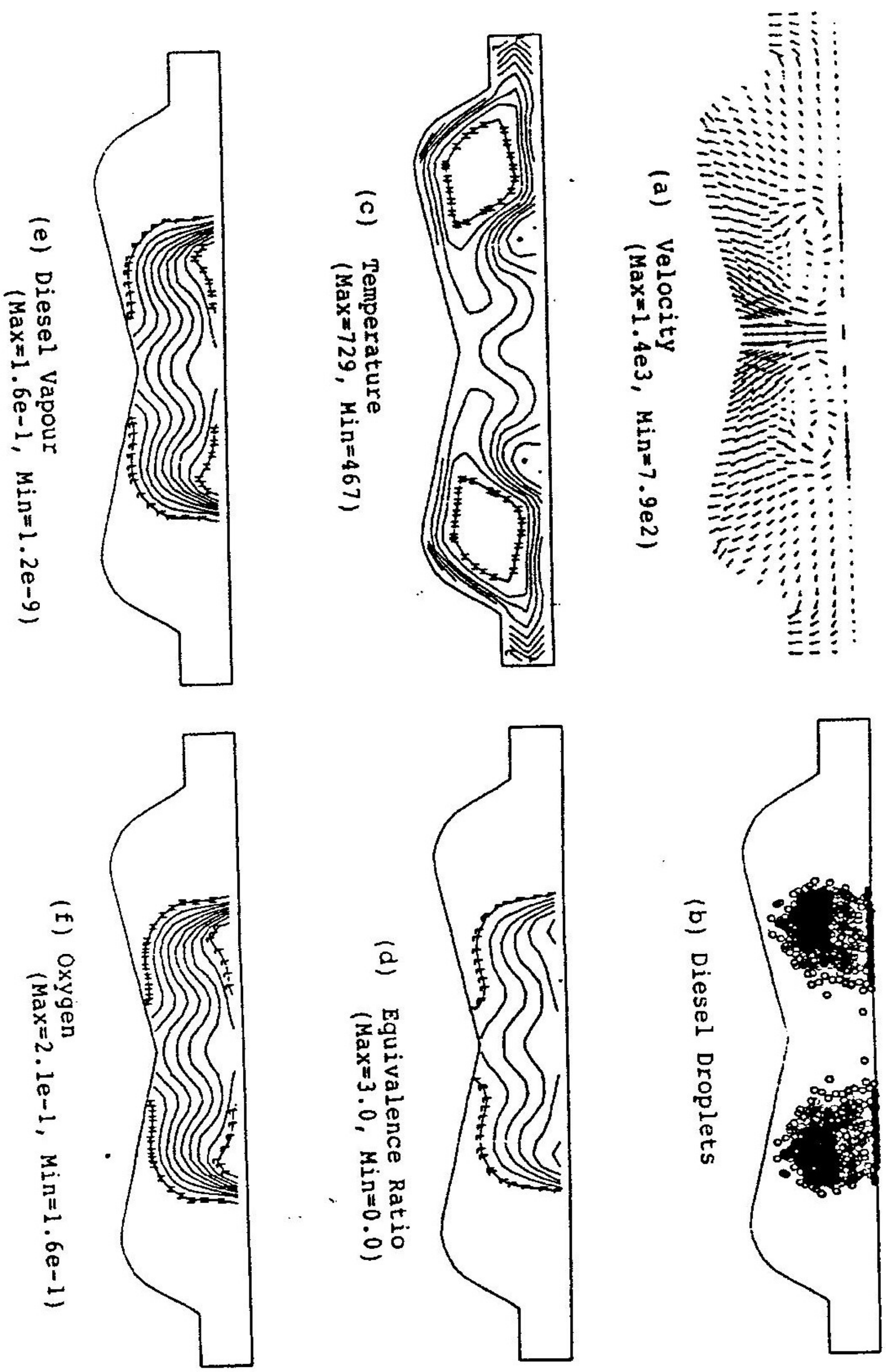

Fig 4: In-cylinder flow processes (20 deg after TDC)

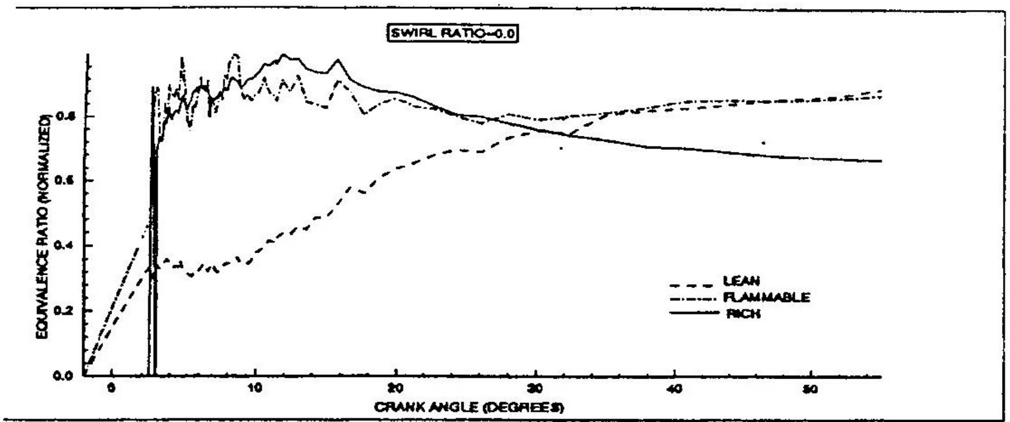

Fig. 5a: Mixture formation at swirlratio of 2.5

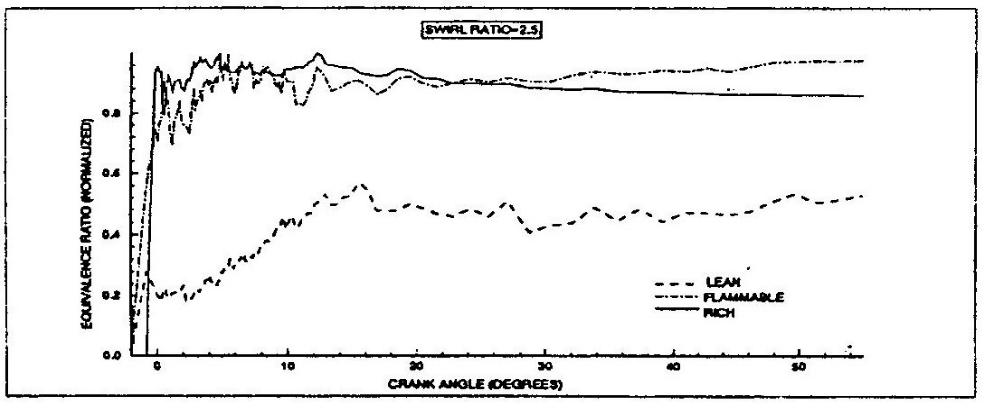

Fig. 5b: Mixture formation at swirlratio of 2.5

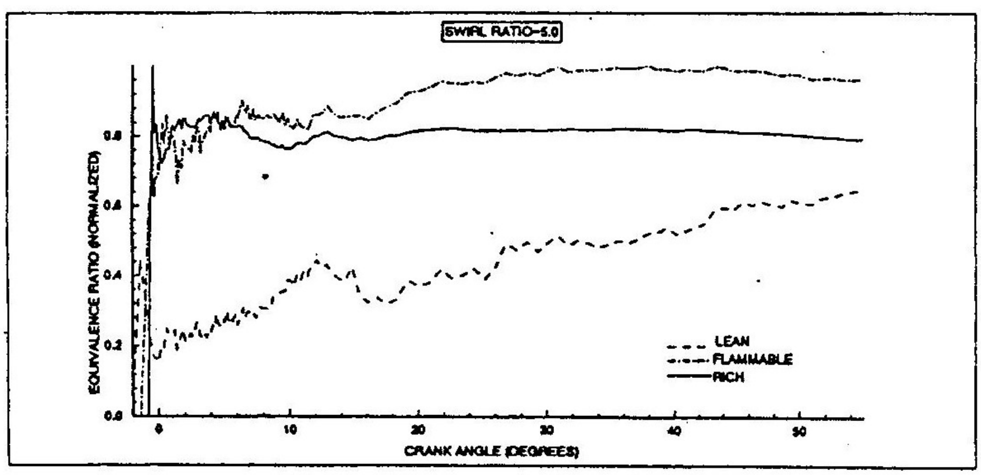

Fig. 5c: Mixture formation at swirlratio of 2.5

that the equivalence ratio at the core of the spray is high due to the absence of free oxygen and that the high mixing intensity takes place mainly at the periphery of the spray where the shear layer is dominant.

Diesel Vapour

Figures 2e through 4e depict the diesel vapour (in grams) existing in the chamber. The amount of diesel vapour increases as time elapses while the chamber temperature is high enough to promote evaporation. It is observed that the vapour is spreading radially with time to seek zones of available air.

Available Air (Oxygen)

Figures 2f to 4f depict the available air (in grams) in the chamber and its location. As it was expected, the local amount of air available is less where amount of fuel is high. This represents a mixing phenomena of utilizing air to form a combustible fuel/air mixture. However, oxygen can not penetrate the rich fuel core since there is no turbulence activities in the core center.

Effect of Swirl

Figures 5a to 5c depict the effect of swirl on the fuel/air mixing. The equivalence ratios in every regime are normalized by the highest value of fmax. Equivalence ratio is calculated from the fuel vapour in the chamber It is interesting to notice that effective fuel/air mixing takes place within the first five degrees (~ 1ms at 1200rpm) after injection. This shows that early part of fuel/air mixing is mainly due to the turbulence generated by the spray itself.

Swirl Ratio of 0.0

From numerical results the ϕ_{max}=3.053 which occurs at about 15 degrees crank angle (CA) after top dead center (ATDC). However, mixing occurs well within the first 5 degrees CA. This indicates that the turbulence generated by the jet is significantly responsible for fuel/air mixing. The lean mixture seems to become fuel-rich approaching the flammable regime.

Swirl ratio of 2.5

Figure 5b illustrates the mixture formation at a medium swirl which has a maximum of ϕ_{max}=2.75. The presence of swirl seem to enhance mixing by lowering the ϕ_{max} closer to the flammable limit. This implies that a significant amount of fuel vapour has formed a combustible mixture. At 5 degrees CA after injection, the flammable mixture seems to remain fairly constant over time because the lean and rich mixtures respectively enriched and leaned to the flammable limits.

Swirl Ratio of 5.0

The maximum ϕ_{max}=3.027 is attained within the first 2 degrees after fuel injection. It drops slightly at TDC and remains relatively constant throughout. This implies that at high swirl level the rich fuel rotates with air (solid body) with less possibility of further mixing. However, the lean mixture is linearly increasing because it is being enriched by the vaporizing diesel.

CONCLUSIONS

From the results it is concluded that:

(i) For the given engine combustion chamber and fuel injection parameters the optimal swirl ratio is between 0 and 5.0. The swirl ratio of 2.5 attained the lowest equivalence ratio for the rich mixture which indicates that substantial amount of fuel was within the combustible range.

(ii) The turbulence generated by the fuel spray jet is very much responsible for fuel/air mixing in combustion chamber, especially at the early part of mixing. Combustible fuel/air mixture was achieved in a quiescent chamber (zero swirl ratio). This suggests that for a low speed running engine swirl is not so much important since the energy carried by the spray jet can enhance fuel/air mixing.

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