Prediction of cylinder pressure in a diesel engine using an improved mixture formation model

K.N.Abdalla

ABSTRACT

A mathematical model for predicting the mixture formation and rate of heat release in a direct injection diesel engine has been improved and modified to take into account the variation of swirl ratio within the combustion chamber. The swirl ratio is found to vary with engine speed and crank angle position from the start of fuel injection to exhaust valve opening. This swirl ratio variation is used to calculate individual zone concentrations which were ultimately needed to solve the heat release model equations.

Results were verified experimentally by comparing predicted and measured cylinder pressure diagrams for different engine conditions. A fairly good agreement between the predicted and measured cylinder pressure was noticed among most of the running conditions.

1. INTRODUCTION

The combustion in a diesel engine is a very complex phenomena in which the processes of fuel atomization, evaporation, mixing and chemical kinetics are inter-related in a relatively unknown manner. These processes superimpose upon and influence each other such that accurate quantitative mathematical descriptions have been impossible to formulate till now (1). It is unavoidable, therefore, that major simplifications must be made to arrive at some general workable relationships for analytical considerations of the heterogeneous burning in diesel engines.

A knowledge of the rate of heat release is necessary in order to predict the behavior of a diesel engine with altered operating conditions by means of cycle simulation, theoretical models. Most of these models are based either on droplet evaporation and combustion or jet mixing phenomena. Models based on droplet evaporation and combustion partially take into account the complex nature of fuel combustion (2). In the jet mixing models (3,4,5) the fuel and air mixing process was thought to be the dominant factor. Due to the complexity of diesel combustion, jet mixing models usually depend on some experimentally adjusted coefficients that relate ignition delay, spray penetration, concentration distribution, etc.

In the work by Meguerdichian and Watson (5) dependance on empirical relations was reduced. Specifically they avoided the use of an ignition delay correlation. Instead they predicted ignition on the basis of the jet mixing process and temperature dependant reaction rates. This model was found very useful in predicting soot release in diesel engines (6). Hence it was chosen for development and improvement in this study. For a detailed description of the model the reader should refer to Meguerdichian and Watson (5) . A brief description is given in the following section.

2. THE MIXTURE FORMATION MODEL

The model is based on consideration of typical fuel sprays formed as a result of fuel injection through a multi-hole nozzle. The fuel spray in its simplest form is considered initially as a free circular jet with a radial wall jet emerging from it. Each spray is divided into a number of zones such that each zone represents the motion of the same fuel element. The detailed zone division patterns have been explained by Meguerdichian and Watson (5). The first law of thermodynamics, the equation of state and the burning rate equation have been applied to individual zones. Pressure is assumed equal in all

* *Agri. Engineering Department King Faisal, University. Saudi Arabia.*

zones at any one time. The fuel mass burning rate is calculated by the Arrhenius Equations, for all zones from the rich to the lean limits of inflammability. These equations, combined together are used to predict the rate of heat release in the combustion chamber.

3. AIR SWIRL

In most diesel combustion simulation models the effect of air swirl has been neglected or rather avoided which is mainly attributed to the difficulties encountered in the theoretical analysis of the diesel engine heterogeneous combustion process.

Air swirl is present in the combustion chamber in the form of solid body rotation about the point of injection and is assumed to confront fuel jets and deflect them accordingly. The angular velocity of the air motion in engines is usually given in terms of swirl ratio (S.R) which is defined as the equivalent solid body RPM of the air motion divided by the engine RPM:

$$S.R. = \frac{RPM(SWIRL)}{RPM(ENGINE)} = \frac{\omega_s}{2\pi(RPM)} \quad (1)$$

where w_s is the swirl angular velocity.

The tangential velocity is given by:

$$u_s = X_s \cdot \omega_s \quad (2)$$

The swirl angular velocity (ω_i) during the induction period is given by Dent (7) as:

$$\omega_i = \left(\frac{8}{D^2}\right)\frac{\int_o^t m.u_s.R_v.dt}{\int_o^t m.dt} \quad (3)$$

During the compression period air swirl is evaluated by applying the principles of conservation of momentum to the contents of the engine cylinder. The swirl angular velocity during compression (ω_c) is calculated as follows :

$$\frac{d}{dt}(I_c \omega_c) = 0 \quad (4)$$

Where I_c is the inertia of the cylinder contents and is calculated from:

$$I_c = \frac{M_t}{2}\left[\frac{V(\theta)+V_b.(D_b/D)^2}{V(\theta)+V_b}\right] \quad (5)$$

At top dead centre (T.D.C), $V(\theta)$ is equal to the clearance volume less the bowl volume which results in a minimum value of I_c and a maximum value of ω. Megerdichian (5) used a constant value of S.R. = 18 for all running conditions. However, in this study the swirl ratio calculations were conducted within a complete cycle simulation program for three engine speeds. The results of these calculations are shown in Fig. 1. The mixture formation program was then modified to take into account this variation of S.R. from the start of fuel injection to exhaust valve opening. the jet deflection parameter, λ is defined by:

$$\lambda = \sqrt{\frac{\rho_a}{\rho_o}\frac{u_s}{u_o}} \quad (6)$$

The parameter, λ is used to describe jet deflection and concentration change in correlations compared to straight jets.

The concentration decay in terms of (s), which is the distance during the curved centerline, is given by:

$$C_{sm} = \left\{\frac{s}{d'} .e^{(7.8\lambda - 1.85)}\right\}^{-1.18} \quad (7)$$

The concentration reduction due to swirl is then calculated for each zone assuming a proportionately factor of the corresponding centerline concentration.

$$C_{st}(K,L) = \frac{C_{sm}(L)}{C_m}.C_s(K,L) \quad (8)$$

The centre line concentration C_{sm} (L) is calculated from equation (7). Zone air entrainment and zone volume are then calculated knowing the zone concentration from equation (8). Individual zone concentrations were implemented for the solution of equations in the heat release model.

4. EXPERIMENAL FASILITY

A Gardner type 1L2, single-cylinder, direct-injection, four - stroke diesel engine was used for experimental verifications. Engine speed ranged from 800 rev/min to 1600 rev/min was measured by using a digital revolution pulse conuter. A commercial optical shaft encoder (Ferranti, type 28, Specification PYD640) was used for angular displaceent measurements.

Cylinder pressure was measured by using an AVL water-cooled piezo-electric pressure transducer flush mounted in the combustion chamber. The signals from the transducer were amplified by using a Kisler charge amplifier prior to feeding them into a data logging system. Values of cylinder pressures were recorded and averaged over fifty engine cycles for each running condition.

Series of tests were performed at naturally aspirated conditions and constant dynamic injection timing for three engine speeds. Test data were obtained at each running condition. These included crank-angle records (averaged over 50 cycles) of cylinder pressure, inlet manifold temperature and pressure, air flow rate, engine speed and torque.

5. DISCUSSION

The mixture formation simulation model described earlier has been developed and improved by the inclusion of the swirl effect in the prediction program. The model was then verified experimentally by comparing predicted and measured cylinder pressure diagrams at naturally aspirated engine conditions for three engine speeds. Predicted (full line) and measured (dotted line) cylinder pressures and fuel burning rates are compared in Fig. 2 toFig. 4. A fairly good agreement between the predicted and measured cylinder pressure is shown in these figures. The predicted fuel burning rates (F.B.R.) are also shown to be in reasonable agreement with those calculated from measured cylinder pressure diagrams. An exact agreement is not expected here because the computation of

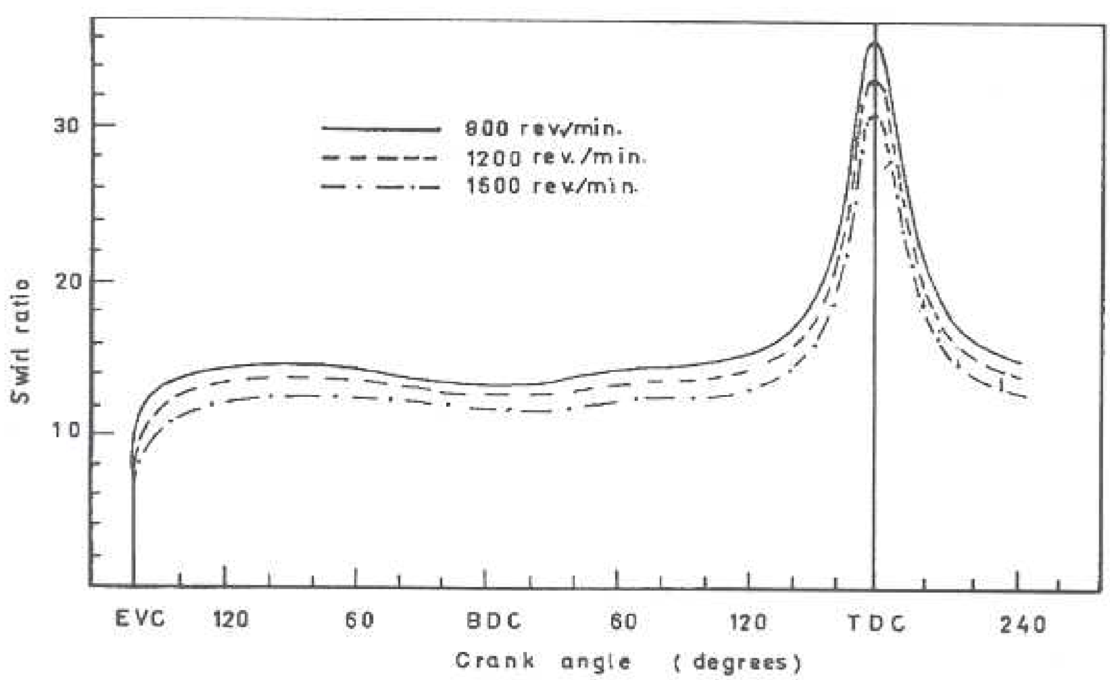

Fig. 1 Variation of swirl ratio with crank angle for different engine speeds

the latter requires the assumption of a homogeneous mixture and requires a uniform temperature in the cylinder at any one time which is not the case in the present model. Nevertheless, the latter is regarded as a useful guideline in various assessments of the predictive capabilities of the model.

The effects of changes in air swirl level (base line, S.R. x0.8, S.R. x1.2) on the prediction of engine performance is shown in fig. 5. An increase in air swirl level is noted to increase the air mass of all zones. Thus at the moment when the mixture first ignites in one zone all other zones approaching their self-ignition temperature contain more air. Increased swirl results in an increase in the initial combustion rate and hence a higher rate of pressure rise is expected. On the other hand, reducing the amount of air in these zones by reducing swirl results in a lower rate of pressure rise and hence a lower peak cylinder pressure as shown in Fig. 5.

6. CONCOUSION

A theoretical multizone mixture formation and heat release model has been developed and improved by including the effect of air swirl during the combustion process. The ability of this model to predict cylinder pressure has been verified experimentally. Reasonably good agreement was obtained in most of the cases considered. The response of the model to changes in air swirl level has been examined and acceptable trends were observed. It is believed that the vast detailed information that this model provides will contribute to the understanding of the diesel engine combustion process.

7. NOTATIONS

C Concentration(fuel/air by mass)

D Cylinder bore diameter

D_b Piston bowl diameter

D_o Nozzle hole diameter

d' Equivalent jet diameter

FCC Fuel/cycle/cylinder

K Layer number

L Zone number in a layer

M mass

m instantaneous mass flow rate

r radius

R_v distance of valve centre from cylinder axis

s arc distance along the deflected jet centre line

$S(\theta)$ distance of piston crown from cylinder head

t time since injection

u velocity

$V(\theta)$ Pistion area x $(S(\theta))$

ω angular velocity

X_s length of the developed part of the jet

y non-dimensional radius

λ deflection parameter

ρ density

Φ_o overall equivalence ratio

SUBSCRIPTS

a air

b burnt

c compression

i induction

o nozzle exit

s swirl

t trapped

z zone

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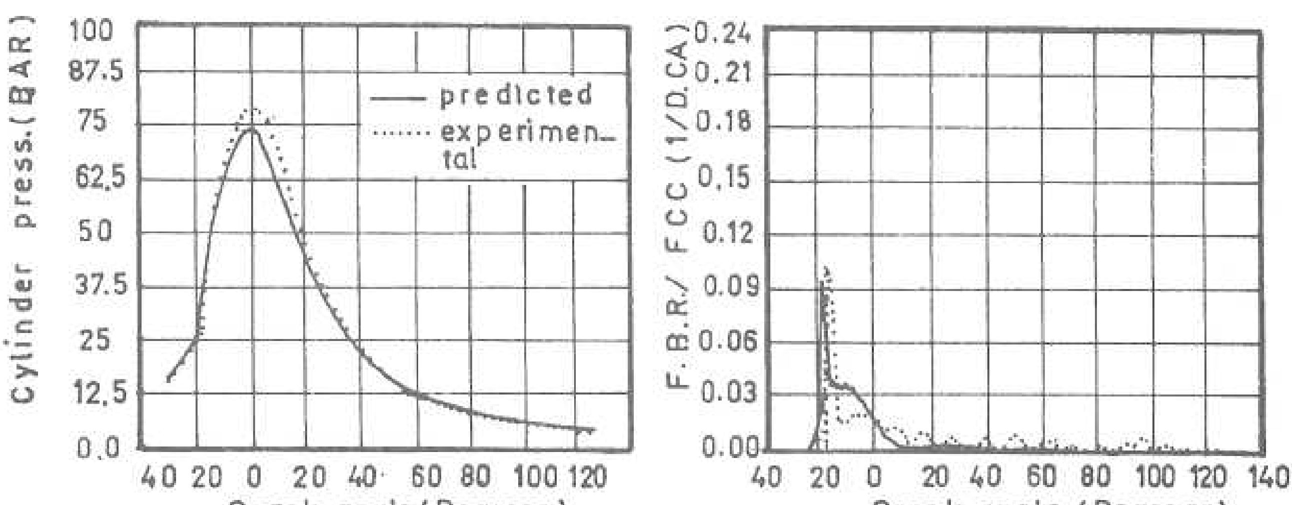

Fig. 2: Predicted and experimental engine performance versus crank angle (N/A engine), speed = 800 rev./min. ϕ_o = 0.82

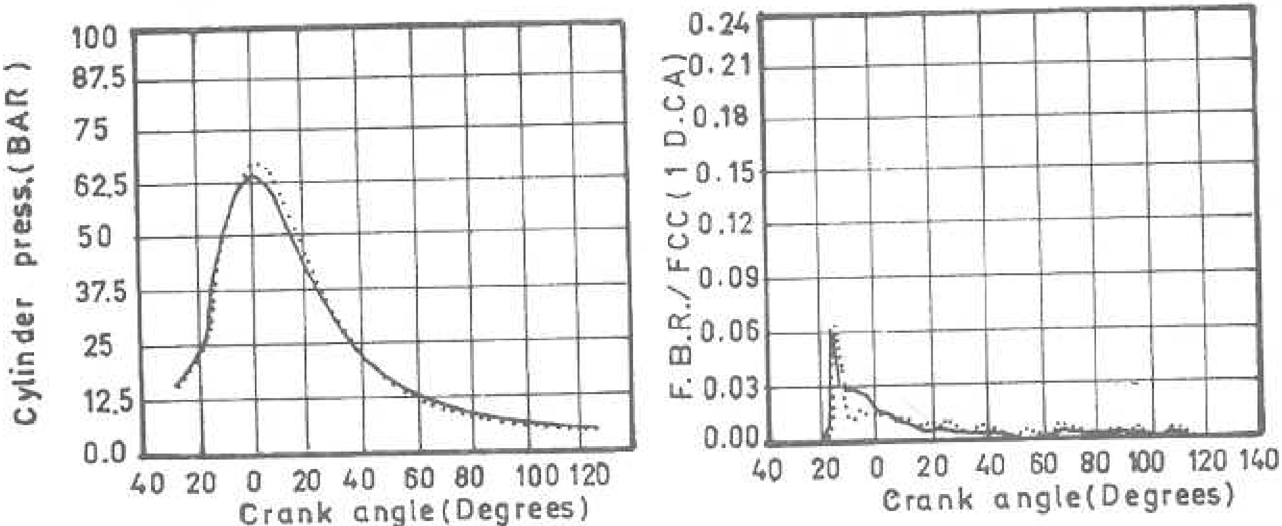

Fig. 3: Predicted and experimental engine performance versus crank angle (N/A engine), speed = 1200 rev/ min, ϕ_o = 0 82

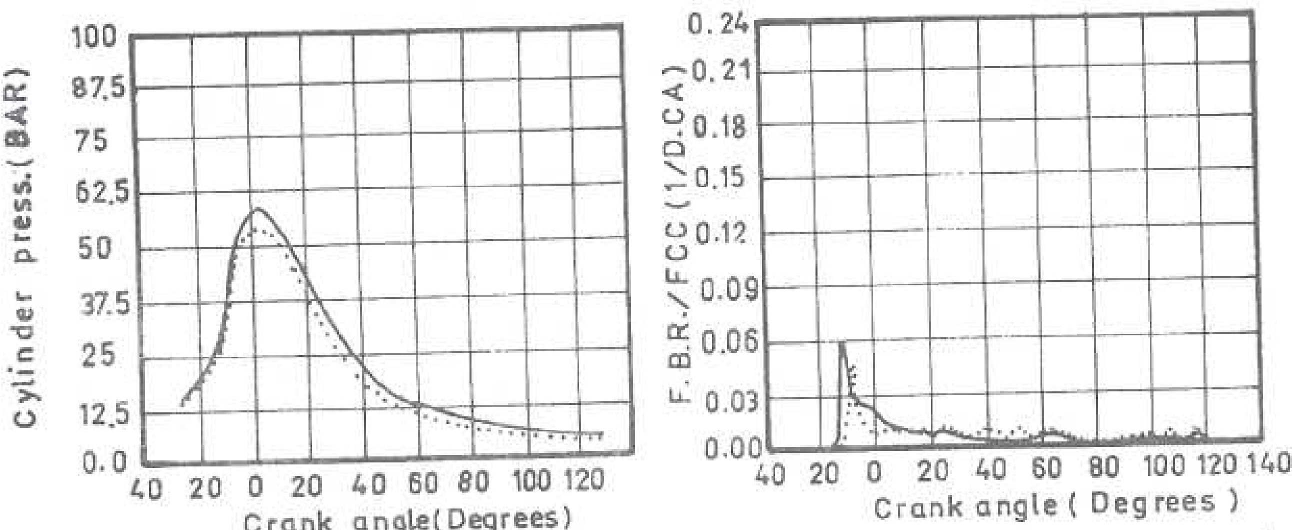

Fig. 4: Predicted and experimental engine performance versus crank angle (N/A engine), speed = 1600 rev/ min, ϕ_o= 0.82

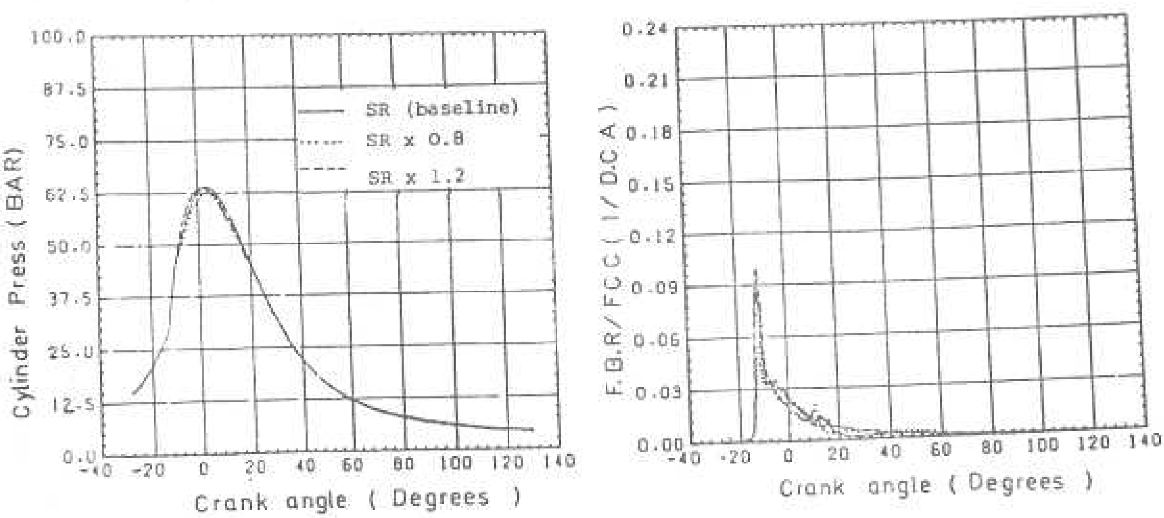

Fig. 5 : Effect of swirl level on predicted engine performance (N/A engine, speed = 1200 rev./ min, ϕ_o = 0.67)