

Study of Fuel Distribution on Diesel PCCI Combustion by Development of a New Characteristic-Time Combustion Model

Yang Yu, Wanhua Su and Haozhong Huang

State Key Laboratory of Engines, Tianjin University

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ABSTRACT

In order to understand premixed charge compression ignition (PCCI) combustion, a new combustion model of kinetic-and-turbulent characteristic-time has been developed. A ununiformity function $H(\phi)$ was presented by analysis of the effect of fuel/air distributions on the role of turbulent timescale in the combustion model, then an analytical turbulent timescale coefficient f was deduced, which was proved to be able to correlate the fuel ununiformity with the turbulent timescale in the combustion model. The new model was employed for simulation of a PCCI combustion organized by various multi-pulse injection strategies in a heavy duty diesel engine. The simulation results agreed with the experimental data well. The ignition process of a PCCI combustion organized by multi-pulse injection was a separated volume autoignition process, which was strongly influenced by the condition of fuel stratification. The combustion temperature of richer zones were drawn down by mass and heat transfer into surrounding lean zones, so that lower combustion temperature and lower formation rate of NO were observed in richer area. Moreover, a “V” type distribution of equivalence ratio was found to be beneficial to retardence of high temperature reaction.

INTRUDUCTION

John Abraham, Frediano Bracco and Rolf Reitz^[1] put forward the laminar-and-turbulent characteristic-time

combustion model for taking into account the effects of both chemical kinetic and turbulence mixing in spark ignition homogeneous combustion. The authors indicated that the early part of flame growth was dominated by laminar effects, so the coefficient of turbulent timescale f was introduced to set the initial value of the turbulent conversion time equal to the laminar conversion time and a turbulent timescale time required for the laminar flame to traverse a distance equal to two or three times the length scale of the turbulent eddies. The model was successfully used for combustion simulation of spark-ignition gasoline engines^[1] and also spray-combustion in diesel engines^[2,3]. The wide range of equivalence ratio and non-homogeneous spray droplet combustion makes the monitoring of flame kernel growth impossible, in which f depends on the appearance of products as an indicator of mixing following the initiation of combustion events with a new formulation^[3].

The chemical kinetics was believed to play a dominant role in combustion process of HCCI, whereas turbulence mixing couldn't be ignored^[4]. Songcharng Kong and Rolf Reitz used a modified f in the diesel combustion simulation to model HCCI with detailed chemistry^[5,6]. The f formulation was used to account for mixing effect by adjusting the model constant.

To consider the discrepancy of characteristic time of local air-fuel ratio, Li Fan and Rolf Reitz^[7] also developed a combustion model for spark-ignition gasoline engines.

Modified f which values were calibrated in the CFD study is to empirically compensate for the over-prediction of the combustion time scale under lean and rich condition, respectively

Kitamura Yasutaka, Kee Sub and Mohammadi Ali^[8] found that the interaction of fuel mixing and chemical kinetics finally determines the feature of combustion process in the experimental research by varying different nozzle orifice diameter, injection pressure and oxygen concentration in a constant-volume vessel.

In the author's previous study^[9-13] it was found that premixed diesel combustion (PCCI) by early multi-pulse injection couldn't avoid fuel and temperature stratification, but characterized much lower overall and local fuel/air equivalence ratio compared with conventional diesel combustion. It was also found that the ignition, heat release and emissions may be influenced by different stratification.

Timothy Jacobs, Bohac Stanislav and Assanis Dennis^[14] found the rich PCCI would result in worse CO and uHC emissions compared with lean PCCI. Flowers Daniel and Aceves Salvador also studied the effect of charge non-uniformity on the emissions under varied equivalence ratio in PCCI engine combustion^[15].

Sjöberg Magnus and Dec John found that by controlling the swirl and two-stage ignition fuels the heat release rates could be controlled so that significantly decreased the peak heat release rate and lowered the knocking intensity^[16,17]. It was also found by Sjöberg Magnus and Dec John that partial fuel stratification offers good potential to achieve a staged combustion event and increase the high load limits. These two stage ignition become sensitive to the fuel distribution because the intensity of the low temperature autoignition reactions is proportional to the local fuel concentration which makes the timing of the "hot" main combustion event sensitive to the local equivalence ratio^[17].

Therefore, it is significant to study the effects of ununiformity of fuel concentration and temperature distribution on the processes of ignition and combustion in

PCCI combustion. The effects of stratification of concentration and temperature are produced finally by the interaction of chemical process and turbulent mixing, especially, for the PCCI combustion organized by early fuel injections. It is physically different from both conventional spark ignition gasoline combustion and conventional direct injection diesel combustion. For SI gasoline combustion, the burning rate depends mainly on chemical reaction rate because of homogeneity of the mixture, even though the flame propagation velocity is influenced by turbulence intensity. For D.I. diesel engine, combustion is dominated by fuel diffusion so that the combustion rate depends mainly on mixing rate. For the PCCI combustion organized by various early fuel injections, the overall fuel/air equivalence ratio is lean, but stratifications of concentration and temperature exist which depend on fuel injection modes and timing. In this paper a new characteristic-time combustion model is put forward in the purpose to simulate the roles of various stratifications of fuel and temperature. A special effort is made to correlate the model with the mixture ununiformity which are prepared by multi-pulse injection strategy proposed in author's previous work.^[9-13]

MODEL FORMULATION

The Shell model is used for auto-ignition kinetics modeling under 1000K^[18] before combustion which is simpler with less computational time than the other ignition models^[19,20]. τ is designated as the overall combustion characteristics-time:

$$\tau = \tau_{ch} + f\tau_t \quad (1)$$

where τ_{ch} is the kinetic characteristics timescale which is the reverse of chemical kinetic rate. Employing Arrhenius overall rate formula, it yields^[4]:

$$1/\tau_{ch} = A[C_{12}H_{30}]^{0.75}[O_2]^{1.5} \exp(-E/RT) \quad (2)$$

where $C_{12}H_{30}$ is taken as a substitute fuel for its heat

value close to diesel oil, $A = 7.68 \times 10^8$ ^[3], $E = 77.3 \text{ kJ/mol}$;

τ_t in equation (1) is the turbulent mixing

characteristics-time, which was expressed as the eddy turnover time, namely:

$$\tau_t = ck / \varepsilon \quad (3)$$

where k is the turbulent kinetic energy, and ε is its dissipation rate, which both are obtained from the RNG $k - \varepsilon$ turbulence model, $c=0.9$.

The weighting coefficient f is designated in this model to weight the role of necessary turbulent mixing in the combustion event. Therefore it is closely correlated with ununiformity of the air and fuel mixture. Basically, Chemical reaction rate depends on the rate of how many fuel molecules meet with molecules of oxygen at a temperature. The more homogeneous the mixture, the less the mixing time. In order to represent the homogeneity of the mixture organized by multi-pulse injection in the cylinder at a certain crank angle, the fuel/air equivalence ratio frequency $g(\phi)$ was used in author's previous work^[21] which manifested the fuel mass fractions at a certain interval of fuel/air equivalence ratio. The equivalence ratio frequency $g(\phi)$ was very directly perceived in graphical analysis, but not convenient in computer fluid dynamics (CFD) modeling. The averaged equivalence ratio $\bar{\phi}$ was further deduced:

$$\bar{\phi} = \frac{\sum_0^{\infty} m_{\phi} \cdot \phi}{\sum_0^{\infty} m_{\phi}} \quad (4)$$

where ϕ is local fuel/air equivalence ratio and m_{ϕ} is local fuel mass at a small interval of equivalence ratio ϕ . It was liberally supposed that more fuel fell with equivalence ratio in the range of $0.5\bar{\phi} < \phi < 1.5\bar{\phi}$, the more homogeneous was the mixture. Based on the supposition an ununiformity function $H(\phi)$ was deduced:

$$H(\phi) = (1 - b \times \sum_{0.5\bar{\phi}}^{1.5\bar{\phi}} g(\phi)) \quad (5)$$

Where b is a coefficient and $b=0.2$. If much fuel with an equivalence ratio ϕ closed to averaged equivalence ratio

$\bar{\phi}$, the mixture was more homogeneous and ununiformity function $H(\phi)$ take lower value. Going further, the weighting coefficient f was expressed as:

$$f = H(\phi) \times \exp\left(\frac{1}{a} \left| \frac{\phi - \bar{\phi}}{\bar{\phi}} \right| \right) \quad (6)$$

The formula (6) makes the weighting coefficient f correlate with the ununiformity of the overall mixture in the cylinder and also with the local feature of uniformity in terms of difference of local equivalence ratio from the averaged value $\bar{\phi}$. The constant a keeps 0.45 in this study.

The time of change of the partial density of species i , due to conversion from one chemical species to another, is given by:

$$dY_i / dt = -(Y_i - Y_i^*) / \tau \quad (7)$$

Where Y_i is the mass fraction of species i , Y_i^* is the local and instantaneous thermodynamic equilibrium value of mass fraction. The characteristics time τ is assumed to be the same for the six species considered necessary to predict thermodynamic equilibrium temperatures accurately: fuel, H₂, CO, CO₂, O₂, H₂O.

The RNG $k - \varepsilon$ model was used for the turbulence calculation^[22] considering the mean strain rate which could predict more realist compressive flow. The nozzle flow model presented by Christopher Sarre, Songcharng Kong and Rolf Reitz^[23] is used to consider the effect of cavitation on the spray atomization in the nozzle. The droplet breakup is modeled by Kelvin-Helmholtz and Rayleigh-Taylor wave hybrid model^[24] based on jet instability analysis. Spray/wall interactions including droplet rebound, splash and film spreading are considered in a wall film submodel^[25]. The integrated numerical

models have been used for diesel engine computations based on the kiva code^[26].

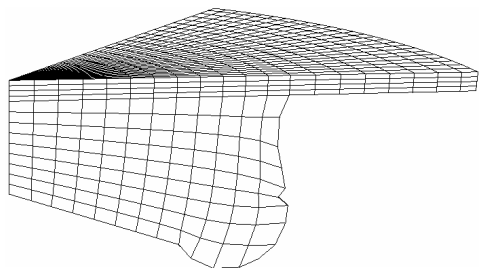


Fig. 1 Computational grid at TDC for the engine

The computations allowed a sector of 51 deg according to one-seventh of the engine combustion chamber, since the injector has seven injector holes. The computations are started at the time of intake valve closure(IVC=-154⁰CA ATDC) and finished at the time of exhaust valve open(EVO=131⁰CA ATDC).The grid cells at TDC is shown in Fig. 1.The surface temperatures of piston, cylinder head and liner are estimated as 500, 450 and 410K, respectively.

ENGINE EXPERIMENTS

The test engine, shown in Fig. 2 is a heavy-duty inline six-cylinder truck engine. In this study, the engine is modified so that the cylinder 1 is used as the test cylinder and the other five cylinders are used as a motor to drive the test cylinder. Table 1 gives the specifications of the engine and the fuel injector. Tables 2 and 3 give the detail of injection modes and experimental conditions. The injection modes are named by number and alphabet. The beginning

number denotes pulse injection number, the alphabet stands for even mode(E),hump mode(H),progressive increase mode(PI).

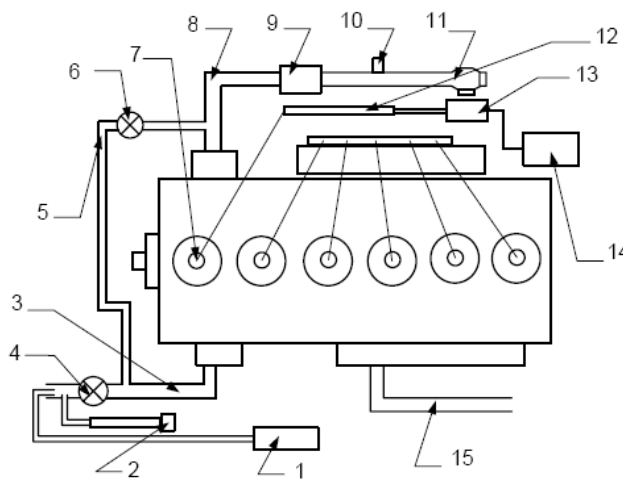


Fig. 2 Schematic of experimental setup

- 1.Exhaust gas analyzer; 2. Bosch smoke meter;3. Single cylinder exhaust port; 4. Pressure controlling valve;5. EGR pipe; 6. Controlling valve;7. Common rail injector; 8. Single cylinder inlet port;9. Air tank; 10. Pressure control valve;11. Air Compressor 12. Common rail;13. Fuel pump; 14. Control system;15. Exhaust pipe

Table 1 Specifications for the test engine and the fuel injector

Combustion chamber	FIRCRI common rail		
	Injection pressure/MPa	Orifice number × diameter/mm	Cone angle/°
BUMP	80	7 × 0.23	155
Bore/mm	Stroke/mm	Compressions rate	Swirl ratio
126	130	15	1.8

Table 2 Test injection mode

Mode	Pulse width of injection (ms) Dwell time						Pulse injected fuel mass (mg) Pulse mass deviation (%)						Total fuel mass (mg)/ Fuel mass deviation (%)
	I	II	III	IV	V	VI	I	II	III	IV	V	VI	
4E	0.864	0.768	0.688	0.688			12.48	9.712	7.997	12.65			42.83/11.58
	0.464	0.48	0.496				12.89	19.39	53.42	36.11			
4PI	0.784	0.832	0.848	0.704			8.634	9.750	16.91	15.94			51.20 / 6.75
	0.416	0.416	0.48				9.557	17.80	16.46	18.96			
5PI	0.672	0.88	0.832	0.704	0.72		4.028	8.248	13.28	10.49	16.48		52.53/20.24
	0.4	0.4	0.448	0.512			25.25	51.89	34.23	52.38	29.50		
6H	0.832	0.8	0.784	0.72	0.512	0.656	9.483	8.708	12.02	14.79	6.725	6.319	58.05 / 9.585
	0.432	0.432	0.48	0.528	0.576		19.79	22.58	31.99	28.58	35.01	36.49	