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Research for the turbulent two-phase flow governing equations in direct-injection engine

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1. Introduction

In the last years a considerable improvement of High Speed Direct Injection (HSDI) Diesel Engine technology has occurred, with a strong increase of fuel economy and a remarkable reduction of emissions and combustion noise [1]. However, more stringent regulations are forcing manufactures of automotives and power plants to reduce pollutant emissions, for the sake of our environment. Turbulent two-phase flow in diesel engine is far from being fully understood; it is probably the most significant unsolved problem in classical physics. Since the flow is turbulent in nearly all engineering applications, the urgent need to resolve engineering problems has led to preliminary solutions based on the Navier-Stokes equations up to a certain point, but then they introduce closure hypothesis that rely on dimensional arguments and require empirical input [2]. This semiempirical nature of turbulence models puts them into the category of an art rather than a science. In the standard KIVA-code, the Taylor Analogy Breakup (TAB) model, originally proposed by Amsden is used to describe droplet breakup. The TAB-model is based on an analogy between a droplet and an oscillating spring-mass system [3]. The external force, the restoring force and the damping forces are analogous to the droplet aerodynamic drag, the liquid surface tension and the liquid viscous forces, respectively. A novel feature of the model is claimed to be its absence of a critical Weber number, for which breakup occurs. Instead, breakup is governed by the droplet oscillation history. A second advantage is that the spray angle is automatically determined, since the product droplets have velocities normal to the respective paths of the parent droplets [4]. However, in applying the model to the spray produced by a high-pressure common-rail system breakup was seen to occur extremely rapidly, which led to a significant under prediction of the liquid and vapor penetration lengths. The TAB-model was found to be appropriate for injection pressure up to around 400 bar, which was also observed by Tanner et al.

In the paper, the governing equations for the turbulent reacting two-phase flow solved in the KIVA-3V fluid dynamics code are presented. An Extended TAB-model (TP) model is proposed and used to calculate cylinder pressure and corresponding heat release rates. One of the features of this model is to equip droplets at the exit nozzle with a deformation velocity such that their lifetime is extended to match experimentally observed jet breakup lengths.

2. Gas phase equations

Applying Favre-averaging and some assumptions to be discussed later, the governing equations for the gas phase can be cast in the following form.

Continuity

$$\frac{\partial \bar{\rho}}{\partial t}+\frac{\partial}{\partial x_i}\left(\bar{\rho}\tilde{v}_j\right)=\dot{\bar{\rho}} \tag{1}$$

Momentum in *i*-direction

$$\frac{\partial\left(\bar{\rho}v_i\right)}{\partial t}+\frac{\partial}{\partial x_j}\left(\bar{\rho}\tilde{v}_i\tilde{v}_j\right)=-\frac{\partial p}{\partial x_i}-\frac{2\partial\bar{\rho}\tilde{k}}{3\partial x_i}\delta_{ij}+\frac{\partial\tilde{\tau}_{ij}}{\partial x_j}+\dot{\bar{F}}_i \tag{2}$$

where

$$\tilde{\tau}_{ij}=u\left[\frac{\partial\tilde{v}_i}{\partial x_j}+\frac{\partial\tilde{v}_j}{\partial x_i}-\frac{2\partial\tilde{v}_k}{3\partial x_k}\delta_{ij}\right] \tag{3}$$

is the Favre-averaging of the viscous stress tensor. δ_{ij} is the Kronecker symbol, δ_{ij} = 1 for $i = j$ and δ_{ij} = 0 for $i \neq j$.

Specific internal energy, *E*

$$\frac{\partial\left(\bar{\rho}\tilde{E}\right)}{\partial t}+\frac{\partial}{\partial x_i}\left(\bar{\rho}\tilde{v}_i\tilde{E}\right)=-p\frac{\partial\tilde{v}_i}{\partial x_i}-\frac{\partial\tilde{J}_i}{\partial x_i}+\rho\tilde{\varepsilon}+\dot{\tilde{Q}} \tag{4}$$

Turbulent kinetic energy, *k*

$$\begin{aligned}&\frac{\partial(\bar{\rho}\tilde{k})}{\partial t}+\frac{\partial}{\partial x_j}\left(\bar{\rho}\tilde{v}_j\tilde{k}\right)=\\&=-\frac{2}{3}\bar{\rho}\tilde{k}\frac{\partial\tilde{v}_j}{\partial x_j}+\tilde{\tau}_{ij}\frac{\partial\tilde{v}_i}{\partial x_j}+\frac{\partial}{\partial x_j}\left(\frac{u}{Pr_k}\frac{\partial\tilde{k}}{\partial x_j}\right)-\rho\tilde{\varepsilon}+\dot{\tilde{W}}\end{aligned} \tag{5}$$

Turbulence dissipation, ε

$$\begin{aligned}&\frac{\partial\left(\bar{\rho}\tilde{\varepsilon}\right)}{\partial t}+\frac{\partial}{\partial x_j}\left(\bar{\rho}\tilde{v}_j\tilde{\varepsilon}\right)=-\left(\frac{2}{3}C_1-C_3\right)\bar{\rho}\tilde{\varepsilon}\frac{\partial\tilde{v}_j}{\partial x_j}+\\&+\frac{\partial}{\partial x_j}\left(\frac{u}{Pr_\varepsilon}\frac{\partial\tilde{\varepsilon}}{\partial x_j}\right)+\frac{\tilde{\varepsilon}}{\tilde{k}}\left(C_1\tilde{\tau}_{ij}\frac{\partial\tilde{v}_i}{\partial x_j}-C_2\bar{\rho}\tilde{\varepsilon}\right)-C_2\rho\tilde{\varepsilon}+C_s\dot{\tilde{W}}\end{aligned} \tag{6}$$

where double indicates denote summation, thus e.g.

$\tilde{\tau}_{ij}\frac{\partial \tilde{v}_i}{\partial x_j}$ corresponds to the tensor product $\tilde{\tau}\nabla\tilde{v}$. $\bar{\rho}$ is the Reynolds-averaged density and $\tilde{v}_i$ is the *i*-th component of the volume-averaged velocity. *P* is the pressure, which, for the cases considered in this thesis, is assumed to be spatially uniform according to the low Mach-number limit. $\tilde{k}$, $\tilde{\varepsilon}$, $\tilde{E}$, $\tilde{J}_i$ are the Favre-average of the turbulent kinetic energy, its dissipation, the specific internal energy and the *i*-th component of the heat flux vector, respectively. $\tilde{\tau}_{ij}$ has been defined above, and *u* is the dynamic viscosity, which has two contributions

$$u = u_{air} + C_u \frac{\tilde{k}^2}{\varepsilon} \tag{7}$$

u_{air} is the laminar contribution and is calculated according to a Sutherland formula

$$u_{air} = \frac{A_1 T^{3/2}}{A_2 + T} \tag{8}$$

where $A_1 = 1.457\times10^{-5}$ and $A_2 = 111$. The second term in the expression for the viscosity is the turbulent eddy viscosity, which is calculated from equations (5) and (6). In flows with high Reynolds number u_{air} is very small compared to the eddy viscosity. C_1, C_2, C_3, Pr_k, Pr_ε are constants resulting from a combination of theoretical considerations and empiricism. Their values for the *k*-ε turbulence model employed in this work are given in Table 1. In addition, the constant C_s is set equal to 1.5. The term $\tilde{\dot{Q}}$ in the energy equation is the source term due to chemical heat release. For the cases studies here, buoyancy effect can be neglected, so no terms containing the gravity have been retained in the equations.

Table 1

k-ε turbulence model constants

C_1	C_2	C_3	Pr_k	Pr_ε
1.44	1.92	-1.0	1.0	1.3

Eqs. (1) and (2) result from the Favre-averaging of the Navier-Stokes equations, which fully describe the fluid flow. As a result of the averaging, unclosed terms appear which require modeling. In order to close the turbulent Reynolds stress tensor in the momentum equations, the so-called Boussinesque-approximation, which is based on an analogy between molecular diffusion and diffusion turbulent eddies, has been employed. It reads

$$-\bar{\rho}\tilde{v}_i\tilde{v}_j = u_t\left(\frac{\partial \tilde{v}_i}{\partial x_j} + \frac{\partial \tilde{v}_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial \tilde{v}_k}{\partial x_k}\right) - \frac{2}{3}\delta_{ij}\bar{\rho}\tilde{k} \tag{9}$$

where u_t is the turbulent eddy viscosity as given by the second term on the right-hand side of Eq. (7). An important implication of introducing a turbulent viscosity of the kind describes here is that the model is incapable of accounting for anisotropy of turbulence. Regarding the small scales, this is not a serious limitation, since they tend to dissipate energy without any directional preference. Large turbulent structures, on the other hand, are clearly anisotropic, which poses a significant constraint for the applied turbulence model.

3. Treatment of the liquid phase

In addition to the solution of the fluid dynamics of the gas phase, modeling the physical processes occurring in a DI diesel engine also requires an adequate treatment of the liquid phase. Current approaches are generally based on a statistical description of the spray. The spray equation [5] describes the evolution of the droplet probability density function (PDF). This PDF has a number of independent variables in addition to time, defined by the amount of statistical information needed. The spray equation has two source terms, one due to droplet breakup (increases the number of droplets) and the other due to droplet coalescence (decreases the number of droplets). The breakup model, which is the subject of the current discussion, should ideally be able to describe primary (atomization) as well as secondary breakup. In the standard KIVA-code, the Taylor Analogy Breakup (TAB) model, originally proposed by Amsden [6] is used to describe secondary droplet breakup. The TAB- model is based on an analogy between a droplet and an oscillating spring-mass system. A novel feature of the model is claimed to be its absence of a critical Weber number, for which breakup occurs. Instead, breakup is governed by the droplet oscillation history. A second advantage is that the spray angle is automatically determined, since the product droplets have velocities normal to the respective paths of the parent droplets. However, in applying the model to the spray produced by a high-pressure common-rail system breakup was seen to occur extremely rapidly, which led to a significant under prediction of the liquid and vapor penetration lengths. The TAB-model was found to be appropriate for injection pressure up to around 400 bar, which was also observed by Tanner et al [7, 8]. An Extended TAB-model (TP) model is proposed and used to calculate cylinder pressure and corresponding heat release rates. One of the features of this model is to equip droplets at the exit nozzle with a deformation velocity such that their lifetime is extended to match experimentally observed jet breakup lengths. Being somewhat related to the TAB-model, the wave-breakup model assumed that breakup is caused by instabilities on the droplet surface. A major advantage of the approach is that the model treats atomization and secondary breakup as indistinguishable processes. Hence, no information of the droplet size distribution at the nozzle exit is needed. Cylindrical blobs with a diameter equal to that of the nozzle exit are injected into the computational domains.

4. Experiment set-up

In the process of engine bench tests, the cylinder pressure must be strictly monitored for the various tanks, therefore, the cylinder head design needs to be modified with the four warm-plugs and the cylinder pressure sensor is installed on the transition casing. Fig. 1 is the cylinder pressure sensor design and structure. The in-cylinder pressure was measured using a flush mounted quartz sensor

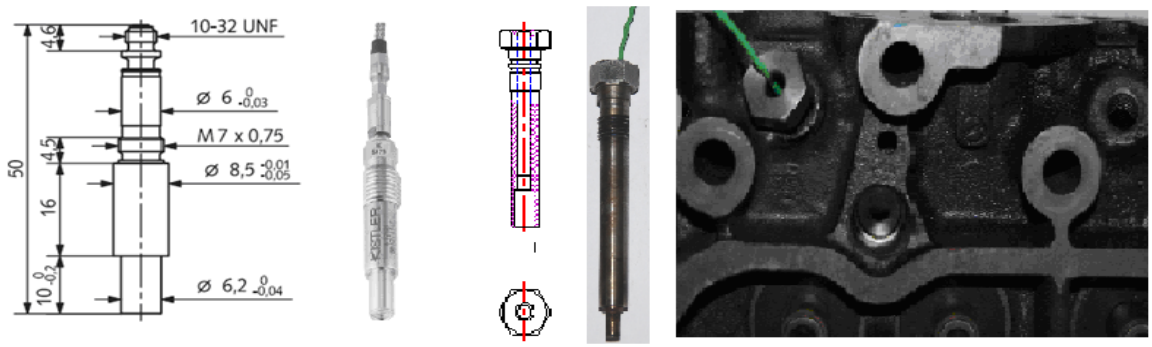
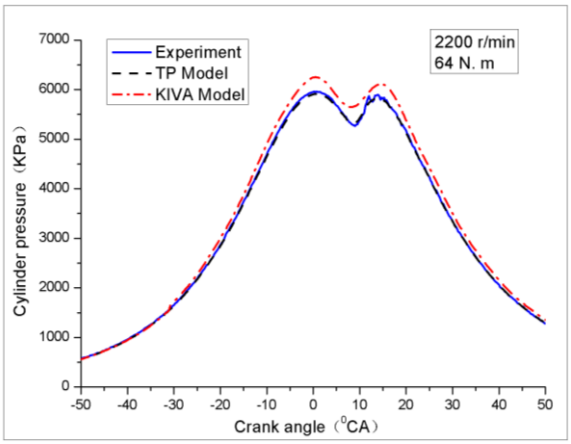


Fig. 1 Cylinder pressure sensor structure and design



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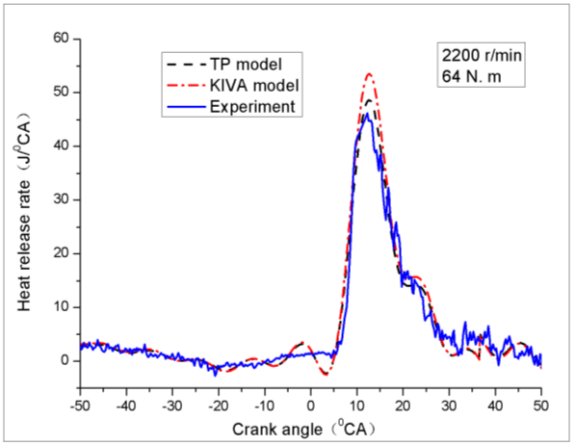

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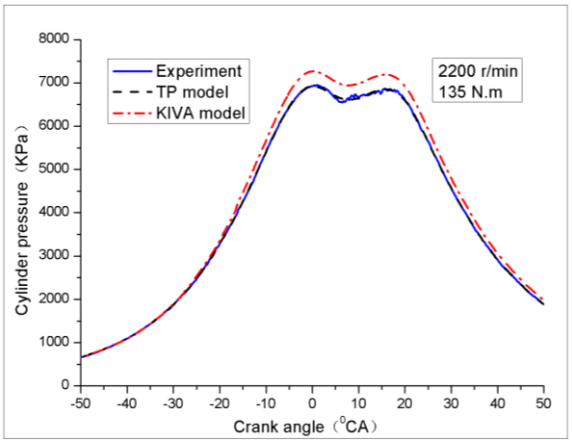

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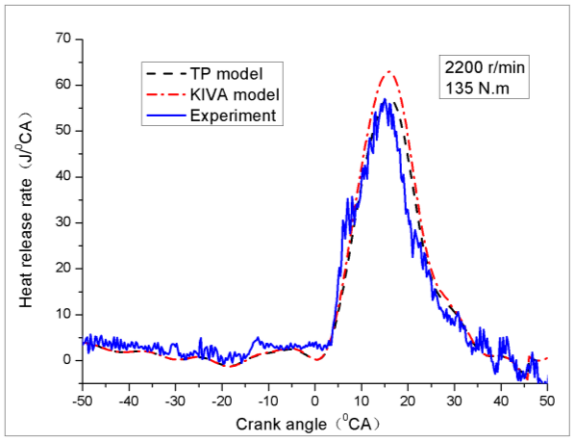

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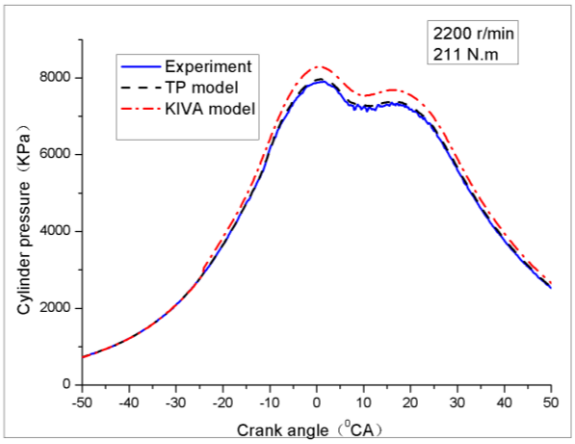

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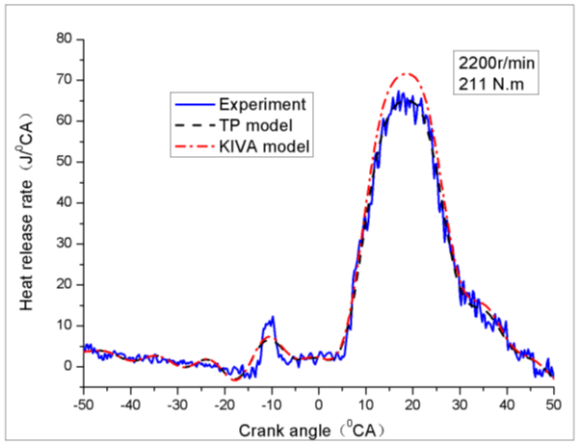

f

Fig. 2 Cylinder pressure and heat release rate: a - cylinder pressure at 2200 r/min, 64 Nm; b - heat release rates at 2200 r/min, 64 Nm; c - cylinder pressure at 2200 r/min, 135 Nm; d - heat release rates at 2200 r/min, 135 Nm; e - cylinder pressure at 2200 r/min, 211 Nm; f - heat release rates at 2200 r/min, 211 Nm

from Kistler (type 6061B) order to assess the performance of the model in terms of predicting cylinder pressure, heat release over a wide range of part load conditions an extensive parameter study varying injection timing, EGR (exhaust gas recirculation)-rate and rail pressure, has bee conducted. First, boundary conditions required for the calculations as well as the operating points investigated are discussed. Next, the basis for the numerical analysis is outlined. Thereafter, the results of the simulations and the measured data are compared. Following this, the results obtained are reflected upon qualitatively in order to assemble a picture of how combustion in a small bore DI diesel engine featuring relatively high swirl proceeds.

The pressure transducer is very accurate on a relative basis but does not directly yield absolute values. In this measurement, the reference was taken to be atmospheric pressure and it had to be corrected for the applied boost pressure. The peak motored cylinder pressure is extremely sensitive to small changes in the pressure at intake valve closing (IVC).

In Table 2 some data on the engine is summarized. The multicylinder equivalent of the investigated single-cylinder test engine is the four cylinders. The injection system is a first generation Bosch Common-Rail featuring a maximum injection pressure of 1350 bar.

Table 2

k-ε turbulence model constants

Displacement	300 cc
Bore	70.0 mm
Stroke	78.0
Connecting tod	132.6 mm
Combustion chamber	Omega-shape bowl
Injector nozzle	6-hole
Nozzle hole diameter	0.124 mm
Injector protrusion	1 mm

5. Results and discussions

For each of the simulations cylinder pressure and heat release rate are compared with measured values. Fig. 2 shows the cylinder pressure traces and corresponding heat release rates for the measured and simulated parameter study varying the work conditions. As can be seen in Fig. 2, the overall agreement in both ignition delay and peak cylinder pressure are excellent. For 2200 r/min 64 Nm, the cylinder pressure is changed quickly when it is 20°CA (Crank angle) BTDC (before top dead center). So the ignition delay time is 12°CA. The biggest experimental pressure is 5800 kPa, which is near to TP model (5820 kPa) and far from KIVA original model (6020 kPa). The heat release rates are similar between TP model (45 J/°CA) and experiment (42 J/°CA), but in the original KIVA model it is 55 J/°CA. This is due to fact the TP model is a multistep chemistry model and KIVA is a one step model. But the difference between the original model and the improved model decreases with the fuel injection quantity, because as the fuel injection quantity increases, power increases, the cylinder temperature and pressure increase and the difference between the multistep and one step models decreases. The cylinder pressures are well fitted between simulation and experiment for all otherwork conditions. The simulation heat release rates are higher than the experiments because of different calculation steps, but the errors are small and the TP model can be used for simulating a real diesel engine.

6. Conclusion

1. The Navier-Stokes equations are closed if turbulent kinetic energy k equation and turbulent dissipation ε equation are added.

2. The calculation applying PDF and experiments of cylinder pressure and the corresponding heat release rates in DI diesel engine are fitted well.

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TURBULENTINIO DVIEJŲ FAZIŲ SRIAUTO TIESIOGINIO ĮPURŠKIMO VARIKLYJE SVARBIAUSIŲJŲ LYGYBIŲ TYRIMAS

R e z i u m ė

Turbulentinio dviejų fazių srauto svarbiausiosioms lygybėms spręsti yra pateiktas išplėstinis TAB (TB) modelis, pritaikytas dujų ir skysčio fazėms. Dujinėje fazėje Favre suvidurkinimas yra panaudotas nenutrūkstamumo momentinei lygybei. Dėl suvidurkinimo atsirado atvirų periodų, kuriuos reikia modeliuoti. Siekiant panaikinti turbulentinį Reinoldso įtempių tenzorių momentinėje lygtyje buvo pritaikytas vadinamasis Businesko supaprastinimas, kuris remiasi molekulių ir turbulentinių srautų difuzijų tarpusavio analogija. Navier-Stake lygtis papildyta turbulentine kinetinės energijos *k* ir turbulentine disipacijos *ε* lygtimis. Diferencijavimo pagal laiką schema kompiuterinėje skysčių dinamikoje (KIVA-3V kodas) įprastai nereikšminga. Lagranžo fazėje, galutinis diferencijavimas taikomas visiems difuzijos dydžiams ir dydžiams, turintiems ryšių su slėgio bangų plitimu. Metodas panašus į pusiau baigtinį metodą, taikomą su slėgiu turinčioms ryšį lygybėms (SIMPLE algoritmas) spręsti jungtinių galutinių lygybių sistemoje. Skystojoje fazėje tikimybinio tankio funkcijos metodas taikomas lašelių išpurškimo lygtims spręsti. Teiloro suirimo analogijos modelis, naudojamas standartiniame KIVA kode, yra išplėstas naudojant deformacijos greitį, kad jį būtų galima pritaikyti eksperimentiškai nustatytam srovės suirimui. Be to, yra aptarti kiti modeliai, kaip antai Rayleigh-Taylor (R-T) modelis, pavienių lašelių modelis ir kt. Apskaičiuotas ir palygintas eksperimentiniu cilindro slėgis ir atitinkamas šilumos atidavimo greitis tiesioginio įpurškimo variklyje. Sukurta teorija ir metodika turbulentiškumui tiksliai apskaičiuoti tiesioginio įpurškimo dyzeliniame variklyje.

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RESEARCH FOR THE TURBULENT TWO-PHASE FLOW GOVERNING EQUATIONS IN DIRECT-INJECTION ENGINE

S u m m a r y

To calculate the governing equations for the turbulent two-phase flow an Extended TAB-model (TP) model is presented to fit for the gas phase and the liquid phase. In the gas phase, Favre-averaging is used for continuity equation, momentum equation. As a result of the averaging, unclosed terms appear which require modeling. In order to close the turbulent Reynolds stress tensor in the momentum equation, the so-called Boussinesque-approximation, which is based on an analogy between molecular diffusion and diffusion turbulent eddies, has been employed. Turbulent kinetic energy *k* equation and turbulent dissipation ε equation are added to close the Navier-Stokes equations. The temporal differencing scheme in CFD(computational fluid dynamics) code (KIVA-3V) is largely implicit. In the Lagragian phase, implicit differencing is used for all diffusion terms and terms associated with pressure wave propagation. A method similar to the Semi-Implicit Method for Pressure Linked Equations (SIMPLE)-algorithm is used to solve the couple implicit set of equations. In the liquid phase, probability density function (PDF) method is used for solving droplet spray equation. Taylor Analogy Breakup (TAB) model used in the standard KIVA-code is extended with a deformation velocity such that their lifetime is extended to match experimentally observed jet breakup lengths. Furthermore, other models such as Rayleigh-Taylor (R-T) model, Discrete Droplet Models (DDM) etc are discussed. Finally, cylinder pressure and corresponding heat release rates in a directed-injection diesel engine are calculated and compared with the experiments. It gives the theory and method about calculating turbulence exactly in directed-injection diesel engine.

Keywords: turbulent two-phase flow, direct-injection engine.

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