Выводы. Получены уточненные зависимости для расчета средних температур теплоносителей в теплообменных аппаратах с перекрестным током и сложной смешанной схемой их движения. Проведенные исследования показали, что традиционный подход приводит к неточностям определения средних температур (см. табл.1 и рис. 4). При больших и малых значениях отношения водяных эквивалентов и больших поверхностях теплообмена погрешности могут достигать до 90%. От точности определения средних температур теплоносителей зависит точность вычисления теплофизических свойств теплоносителей и материалов поверхности, коэффициентов теплоотдачи и теплопередачи, средних температурных напоров и температур стенок. Представленные зависимости позволяют более точно рассчитывать процессы теплопередачи в теплообменных аппаратах и являются базовыми для проектных, поверочных и оптимизационных расчетов.

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EXPERIMENTAL INVESTIGATION AND MODELLING OF DIESEL ENGINE FUEL SPRAY

Introduction

The optimization of diesel engine fuel spray is one of the ways for improvement of engine efficiency and soot reduction. A research programme on spray characterization by laser diagnostics has been carried out at Sir Harry Ricardo automotive centre, University of Brighton, UK. Transient penetration of a Diesel spray has been explored when fuel is injected into quiescent air in *Proteus* combustion chamber [2,3,4,5]. The spray penetration modelling has been discussed by a number of authors [6, 7, 8, 10]. Traditionally CFD simulation (e.g KIVA) is used for multidimensional spray modelling [4, 9]. Under Lagrangian-Eulerian approach the spray is modelled as an ensemble of droplet parcels. Each parcel is characterised by its own droplet size, temperature and specified injection velocity. In some cases however the prediction can be worse than that by an empirical correlations or a simpler spray model [5]. This can be attributed to the intrinsic deficiency of the Lagrangian-Eulerian approach for dense sprays. An underlying assumption of this approach is that liquid fraction is small when compared with gas phase fraction. Clearly this is not true in the vicinity of a diesel nozzle.

Hence there is a need in a simple and robust approach to spray penetration modelling from the first principles (conservation of momentum). This is suggested by the COFM model as described below. Another viable alternative is given by the popular DIESEL-RK (Bauman MSTU) simulation based on Razleytsev model [1].

In the proposed COFM model [4, 5] the conservation of momentum is applied to the whole spray as a physical body. In other words, the main difference between the proposed COFM model and the Lagrangian-Eulerian approach is that the conservation of momentum is applied to whole spray for the former rather than to separate droplet parcels for the latter.

The experimental mass flow rate is used as an input data into the COFM model, and the model predictions are validated against laser diagnostics data. The experimental setup is described below.

Experimental apparatus and procedures

The experimental apparatus is a rapid compression machine based on Ricardo *Proteus* single cylinder two-stroke test engine with a specially designed head for optical access for spray visualisation. The spray chamber within the optical head has a cylindrical shape of radius 25 mm and 80 mm in height; this prevents any wall impingement for the spray [5].

A second generation common rail system was used with a maximum injection pressure of 160 MPa. The injectors used for this study were a valve covered orifice (VCO) nozzle with a 3 holes (diameter of 0.2 mm, Bosch) and a 7 holes nozzle (diameter of 0.135 mm, Delphi).

To characterise the liquid spray penetration, a Phantom V7.1 high-speed video camera was used. The processing of the video images for measurements of the spray penetration was performed by purpose developed software [2].

Fig.1 shows measured spray tip penetration at different injection pressures for Bosch and Delphi injectors. It should be kept in mind that the optical access into *Proteus* window is possible only up to 45 mm of spray penetration.

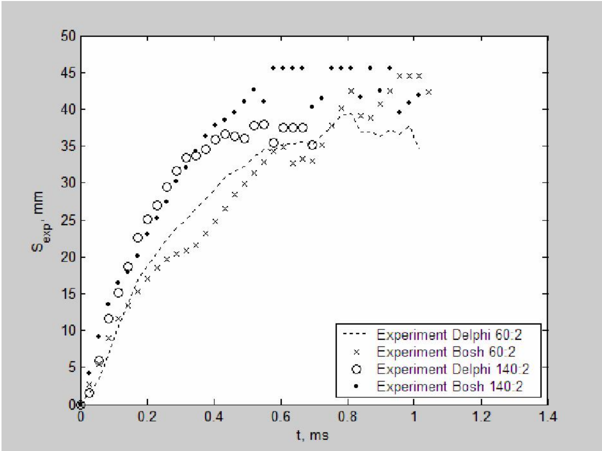


Fig. 1. Comparison between Bosch and Delphi spray tip penetration experiments; ambient pressure 2 MPa; injection pressure 60 MPa and 140 MPa

A long-tube technique was used to measure the instantaneous mass flow rate. Liquid fuel was injected into a long tube containing the same working fluid under pressure [3]. A pair of strain gauges measuring variation in the internal pressure is fitted to the tube immediately downstream of the injector nozzle [4].

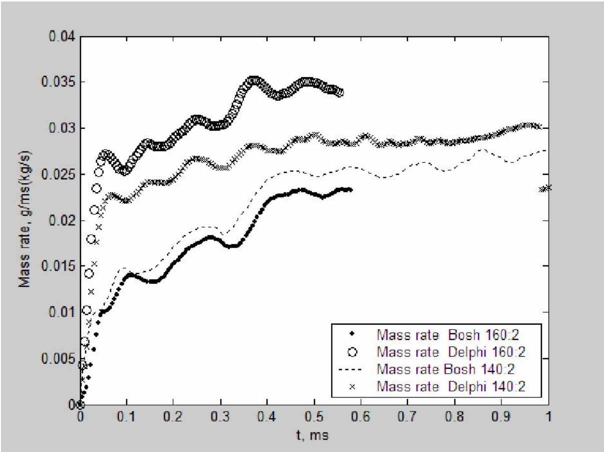
Fig. 2. shows the measured mass rate at different injection pressures for Delphi and Bosch injectors.

COFM model for spray penetration of diesel fuel

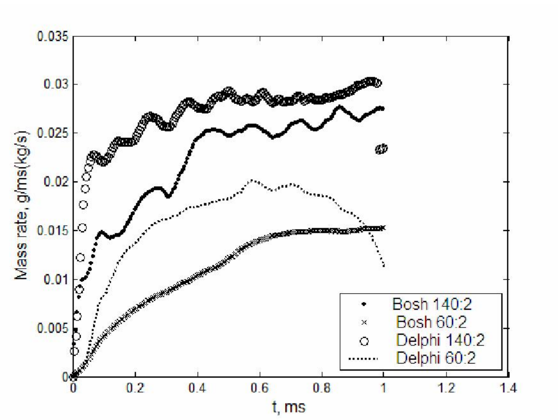
The equations for momentum conservation for the spray can be written as [4, 5]:

$$\frac{d(mu)}{dt} = \rho_l \; A_n \; u_{inj}^2(t) - \frac{1}{2} C_D \rho_g \; A(t) \, \beta^2 u^2 \qquad (1)$$

where m is injected mass; t is time from the start of injection; u is velocity of centre mass of the spray; ρ_1 is liquid fuel density; A_n is nozzle (hole) exit area; $u_{inj}(t)$ is instantaneous injection velocity; c_D is drag spray coefficient; ρ_g is ambient gas density; *A(t)* is frontal spray area; β is the ratio of spray tip penetration to the position of centre of mass.



a



b

Fig. 2. Mass rate for Delphi and Bosch injectors; ambient pressure 2 MPa; a – injection pressure 140 MPa and 160 MPa; b – injection pressure 60 MPa and 140 MPa

In this study the drag spray coefficient C_D =1.54.

Instantaneous injection velocity $u_{inj}=u_{inj}(t)$ was defined from transient experimental mass profile $\dot{m}$ as:

$$u_{inj} = \frac{\dot{m}}{\rho_l A_n n} \quad (2)$$

where $\dot{m}$ is mass flow rate of fuel injection; n is number of injector nozzles.

As it was shown based on the experimental data [4,5], the spray tip penetration S_{tip} can be linked with good accuracy to the position of centre-of-mass $s(t)$ as $s_{tip}(t) = \beta s(t)$ where β is a constant in the range 1.3 to 1.9.

In the present model it is assumed that the shape of the injection spray until cluster shedding is a cone with the height S_{tip} and frontal spray area $A(t)$. The expression of the volume of the spray cone can be written as

$$V(t) = \frac{1}{3} A(t)\beta s(t), \quad (3)$$

where $S(t)$ is the spray penetration based of distance travelled by the centre of mass of the spray (COFM).

The velocity of COFM of the spray is:

$$u = \frac{dS}{dt}, \quad (4)$$

where $S = S(t)$.

The volume occupied by the spray consists of injected fuel and entrained air. The liquid fraction of the spray is defined as the ratio of liquid volume in the spray to the total volume of the spray:

$$\varepsilon = \frac{m(t)}{\rho_l V(t)}, \quad (5)$$

where ε is the liquid fraction in the spray, $m(t)$ is injected fuel mass; $V(t)$ is the volume of the spray; this is including both injected fuel and entrained air.

Hence the frontal spray area can be estimated from (3) and (5) as

$$A(t) = \frac{3}{\beta s(t)} \frac{m(t)}{\rho_l \varepsilon}. \quad (6)$$

It was suggested [4] that the decrease of liquid fraction in the spray due to air entrainment can be written as

$$\varepsilon(t) = \varepsilon_0 \exp(-\frac{t + t_{OD}}{\tau}), \quad (7)$$

where $\varepsilon_0 = 1$ is the liquid fraction in the spray before the commencement of injection process; t_{OD} is injector delay time, τ is the characteristic dispersion time.

The model aims to predict spray penetration from the experimental mass flow rate profile, with the adjustable model parameter, the dispersion time τ. The injector delay time is estimated in the experiment.

The equations (1), (2), (4), (6), (7) can be rewritten as a system of two differential equations with two variables: velocity (*u*) and spray penetration of COFM (*S*):

$$du/dt = A_1 \times u^2 / S + A_2 \times u + A_3 \qquad (8)$$
$$dS/dt = u$$

where A_1, A_2, A_3- coefficients depending on time.

System (8) was solved based on an explicit Runge-Kutta method using bespoken COFM program in Matlab 6.5 [6]. The input data of program are ρ_1 = 800 kg/m^3, ρ_g = 19.9 kg/m^3; nozzle exit diameter D_n = 0.2 mm, n= 3, (Bosch); D_n = 0.135 mm, n = 7 (Delphi).

Model validation against experiments

Analyses of experimental data show that spray penetration between Bosch and Delphi injectors are rather close in the initial stage of Fig 1. The same tendency is observed in Fig 3 which shows the results of calculations using the COFM model (8).

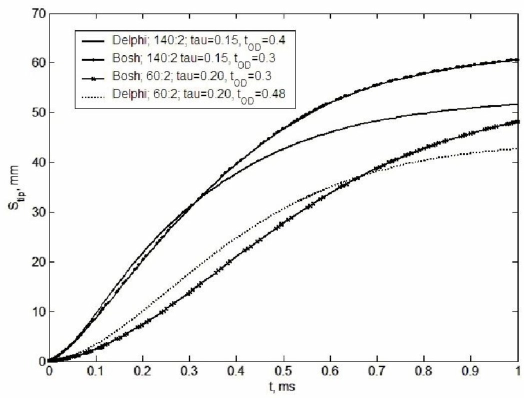
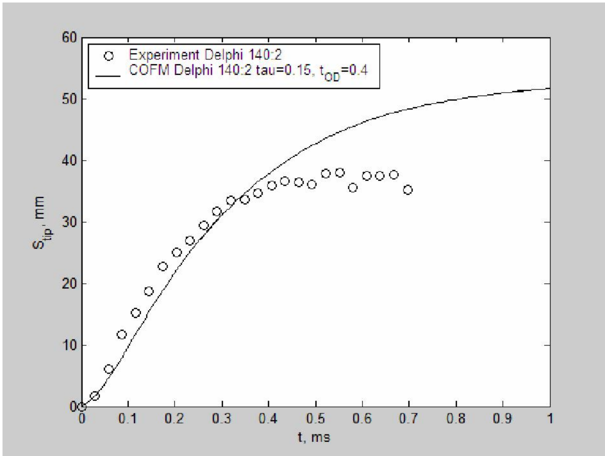


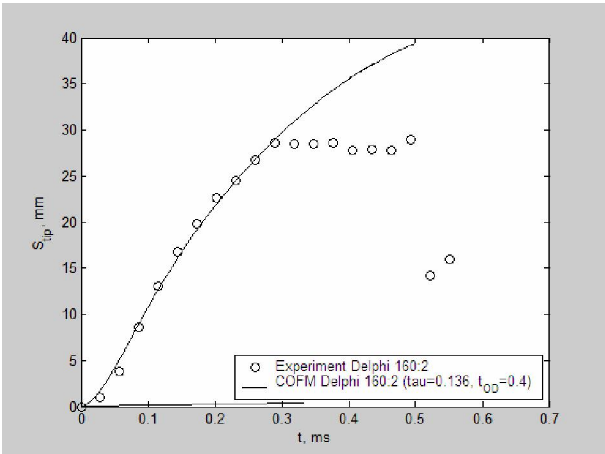
Fig. 3. Spray tip penetration for Delphi and Bosch injectors; ambient pressure 2 MPa; injection pressure 60 MPa (β=1.6) and 140 MPa (β =1.3)

The COFM model is shown to produce a good agreement with the experimental data for spray penetration until major spray instability (cluster shedding [5]) occurs.

Fig. 4. shows the comparison between the measured and predicted spray tip penetration at injection pressures 140 and 160 MPa.



(a)



(b)

Fig. 4. Spray tip penetration for Delphi injector; ambient pressure 2 MPa (β =1.3);
a) injection pressure 140 MPa , tau=0.15 ms;
b) injection pressure 160 MPa , tau=0.136 ms

As it was found by fitting the model predictions to the experimental data on spray penetration, the dispersion time τ is increasing when injection pressure is decreasing. For ambient pressure 2 MPa, the tuning parameter τ is equal to 0.138, 0.15, and 0.20 ms for injection pressure 160, 140 and 60 MPa correspondingly.

For Bosch injector, the same values of τ can be accepted within experimental accuracy. It should be kept in mind that the parameter β was assessed from Delphi data but it is used for Bosch data for an approximate study.

It is relevant to try to link the observed tendency in spray dispersion time τ with the conventional concept of spray breakup time; this will be done in the following section.

Empirical correlations of breakup time

Traditionally breakup time (the time when liquid core breaks up) is assessed by the correlation proposed by Hiroyasu (Heywood, p 530):

$$t_b = 29 D_n \rho_l / \left(\Delta P \rho_g\right)^{0.5}, \quad (9)$$

where ΔP is pressure difference.

It gives breakup time of 0.06 ms in case of Delphi 160:2 and 0.08 ms in case of Bosch 160:2 (injection pressure of 160 MPa, ambient pressure of 2 MPa).

COFM model has the characteristic time for exponential decay of liquid fraction τ as an adjustable parameter. Although the concepts of dispersion time τ and breakup time are defined in a different way, a qualitative correlation between them can be expected. Indeed they show the same trends in the dependence on injection pressure. Breakup time decreases with growth of injection pressure as can be seen from Eqn. (9). The same trend is observed in the values of dispersion times obtained by fitting the model predictions to the experimental data. Summarising it, the dispersion time is τ=0.138 for 160 MPa injection pressure; it is 0.15 ms for 140 MPa and 0.2 ms for 60 MPa.

The COFM model is based on the first principles (momentum conservation) being applied to experimental mass flow rate input data, whilst the Hiroyasu correlation is an empirical one. A qualitative agreement between these two different approaches is encouraging.

Conclusions

1. Diesel spray penetration has been investigated experimentally and modelled theoretically using the COFM model for injecting pressures 60, 140 and 160 MPa and ambient pressure 2 MPa for Bosch and Delphi injectors.

2. The COFM model is shown to produce a reasonable agreement with the experimental data for the initial stage of penetration until spray instability (cluster shedding).

3. For the cases under consideration τ was equal 0.138, 0.15 and 0.20 ms at injection pressure of 160, 140 and 60 MPa correspondingly, for both Delphi and Bosch injectors. The qualitative dependency of τ on injection pressure is close to that reported in the literature for spray breakup times.

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