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Mathematical Modelling and Computer Simulation of Medium Size Diesel Engine Running on Varying Quality Fuels

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ABSTRACT

This paper gives detailed accounts of two different methods of combustion modelling. One is based on jet mixing approach, taking account of the contraction of the spray cone angle and over penetration tendencies of heavier fuels. In this model the delay period is calculated using empirical data.

In the second method, a droplet burning approach is used as the basis for the combustion calculations.

The jet mixing model is supported by two other mathematical models viz fuel injection and heat release. The engine performance parameters in this case are determined using a modified air cycle adopting simplified 'step by step' calculations.

The paper is concluded with a presentation of a series of comparisons between predicted and actual experimental data involving different quiescent medium size diesel engines running under different conditions, burning varying quality fuels. Special references to the modelling approaches adopted are made and their application in engines running on heavy fuels are discussed.

INTRODUCTION

Most combustion models are based on two distinct approaches, "jet mixing" and "droplet burning". While it can be generally concluded that for most of the combustion period the rate of mixing is controlled more by the entrainment of air into the jet, nevertheless, to be able to recognise the different burning behaviour of varying quality fuels and to predict the period of pre-mixed combustion particularly the initial rate of pressure rise, data on droplet size, distribution, relative motion and evapouration rate are required. Furthermore, the distinction between the droplets and outer envelope of fuel/air cloud mixture in the fuel spray is considered important. The interaction of sprays as free or wall jet is sufficiently an important area for consideration.

Historically there has been good reasons for adopting two methods of simulating combustion processes based on the following assumptions respectively:-

1. An assumption that the fuel vapourises very quickly, in which case the fuel is considered to be a gas jet spreading into the surrounding air - the "jet mixing" model.

2. An assumption that the behaviour of independent droplet evapouration and burning is central to the overall combustion processes - the "droplet burning" model.

Neither is considered adequate because;

(i) the jet model does not allow for different evapouration rates of different quality fuels, and

(ii) the jet comprises a cloud of droplets sweeping along in a mass of entrained air as distinct from isolated droplets.

ENGINE/COMBUSTION MODEL

The engine cycle is presented in Fig 1. The engine combustion model is presented in Fig 2. Fig 3 gives a representation of the multi-zone elemental free air and wall jet.

Approach 1. In this approach the spray modelling as it stands does not include the effect of droplets and it is limited to three distinctive zones viz fuel, product and air zones. Considering the actual injection rate diagrams and fuel line oscilograms for typical medium/large diesel engines, it is apparent that fuel elements are injected at different pressures and rates. To this end, work is now being conducted to transform the elemental free air and wall jets as presented in Fig 3 into an 'infinite zone Jet' model, where each half element of fuel injected contains a particular size droplet (1).

Furthermore, as far as the delay period

is concerned, this, in the past has been entered as a known quantity. However, a correlation for this as a function of pressure, temperature and fuel quality (eg CCAI) would be preferable. A correlation has emerged from recent work. (Appendix 2).

Approach 2. From an estimate of the trapped mass of air, the cylinder pressure and temperature can be calculated up to the point of injection. Heat loss through the walls of the combustion space is calculated using the Woschni method (2).

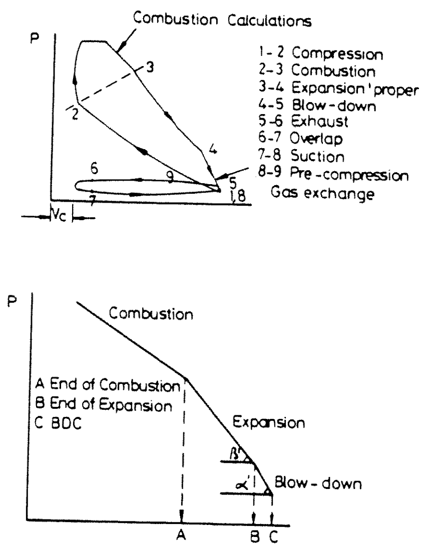
Fuel enters as a jet of droplets which entrains air and spreads into the cylinder as a jet. Penetration of the jet is to be calculated according to the equations of Hiroyasu (3). The jet will be divided radially into five conical shells, chosen such that an equal mass of fuel is contained in each shell.

From the fuel injection pressure diagram the mean droplet size and the mass of fuel entering each axial element can be calculated.

The radial width of the jet is determined from conservation of momentum. This leads to a constant jet angle, independent of charge density. The jet angle is, however, adjustable by varying the radial distribution of velocity. The distribution has shown to give an angle which correlates well with published data.

DIESEL ENGINE SIMULATION - THEORY

The final proposed diesel engine simulation model incorporates a combination of approaches 1 and 2 described earlier for combustion calculations. It is assumed that air entrainment is controlled by an elemental



ne (expansion index) = tan β'
nb (blow-down index) = tan α'

Fig.1 Details of Engine Cycle

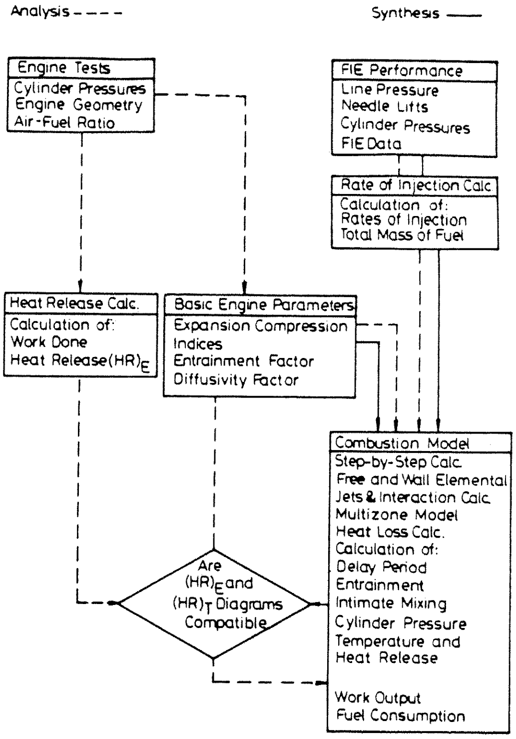


Fig.2 Engine Combustion Model

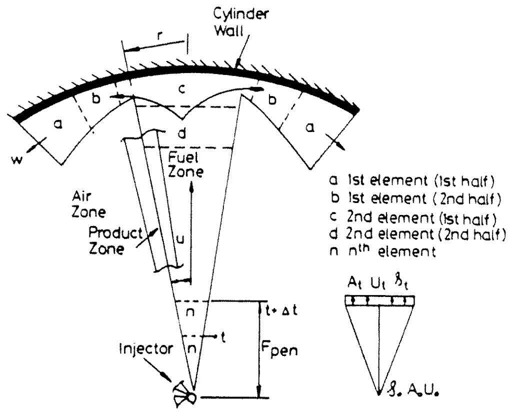


Fig.3 Simplified Presentation of Elemental Free and Wall Jets with No Swirl

jet and after infringement by an elemental wall jet. The intimate fuel air mixing within the jet is controlled by turbulent diffusion; the jet interaction is taken into consideration as presented in Appendix 1.

Pre-Compression Consideration and Compression Period (Process 1 - 2). The compression is basically the first sequence of closed period calculations. Starting from initially assumed trapped conditions at 1, P_2 and T_2 are calculated.

The initial trapped conditions are subsequently modified by the inclusion of clearance volume scavenging calculations.

Different methods of calculation are available viz, simplified perfect displacement model ie

$$M_T = (S.P_A\eta_v + V_{cL}.T_1/T_{exh}).P_1/(R.T_1)$$

or simplified perfect mixing model ie

$$M_T = (S_{eff}.P_A + V_{cL}).P_1/(R.T_1)$$

Once the trapped mass is evaluated, the inlet and the temperature at the start of injection can be obtained.

Applying the momentum equation the overall mean injection pressure is calculated as follows:

$$\Delta P = 855.94.10^{-6}\ [(M_f.RPM)/(h_n.T_I.d_n)]^2$$

$$\text{where, } M_f = \dot{Q}.\rho_f.T_I/(6.RPM)$$

Evaluation of temperature values allows the internal energy of the compressed air to be estimated from the Gilchrist table and the gas constant function. Interpolation and extrapolation is possible but only within the air fuel ratio range given.

The delay period is included in the compression calculations. It is either determined from the cylinder pressure diagram or is calculated. (Appendix 2).

<u>Combustion.</u> The combustion calculation is divided into two periods

i) 'pre-mixed' combustion (chemically controlled and where droplet consideration predomindates).

ii) 'diffusion' combustion (mixing rate controlled and when burning is primarily based on mixing rate).

<u>Spray Jet Calculation (Process 2 - 3)</u> The expression used for mixing is based on modified penetration rate given in (5).

A number of expressions (6-10) were evaluated (Fig 4). The penetration rate given by (5) was satisfactory for all medium sized engines (See Appendix 4) . Fig 5 gives the comparison between experimental and predicted results for engine (B).

For heavy fuels, however with both approaches, it is necessary to depict a modified formulae for cone angle and spray penetration. The research work in heavier fuel behaviour has indicated that the penetration increases and the cone angle decreases as fuel viscosity is increased, Figs 6 to 9. Fuels tested are given in Appendix 3.

The fuel injected first forms a free jet which after inpingement on the cylinder wall is transformed into a wall jet for $t > t_f$ where $t_f = 22.4\ \rho_f dn/\ (\rho_c.\Delta P)$. The centre-line penetration is defined by,

$$F_{pen}=2.2\times10^{-3}(\rho_f/\rho_c)(u_t d_t t)^{0.5}(CCAI)^{1.05}$$

And for $t < t_f$ as given in (3) by,

$$F_{pen} = 17.89\times10^{-5}\sqrt{(2\ \Delta P/\rho_f)}.t.(CCAI)^{1.15}$$

The actual injection pressures are used for cases where data was available. For qualitative analyses mean injection pressure considered satisfactory (Fig 10).

The cone angle can be determined by consideration of the momentum at the point of injection and at any station along the spray path as shown in Fig 3.

$$\rho_t A_t U_t^2 = C_d\ \rho_o A_o U_o^2$$

Where $U_t = df_{pen}/dt$

and $U_o = C_d\sqrt{(2P_j/\rho_0)}$

Leading to

$$\tan\theta = K\ (\rho_c/\rho_a)^{\frac{1}{2}}$$

Where K is constant and can be determined.

Hence the mass entrained is given by,

$$E=\int_t^{t+\Delta t} (\pi/3).\tan\theta.F_{pen}.\ a$$

Where ρa is the gas density.

<u>Wall Jet</u>. Once t is greater than t_{fw}, the jet front changes from a free jet to a wall jet. Transition time, air entrainment during transition and loss of kinetic energy in the direction of flow for the transition are neglected. The initial condition for the wall jet is given by:

$$Y_0 = F\ \tan\theta$$

Where Y_0 is initial wall jet radius.

Assuming the velocities and flow areas immediately before and immediately after the transition are the same, the initial wall thickness and initial wall jet velocity is represented by:

$$S_0 = F/_2\ \tan\theta \text{ and}$$

$$W_0 = dF_{pen/dt} \text{ at } F_{pen} = F$$

The initial volume flow at wall jet can then be found:

$$Q_0 = \pi F^2\ \tan^2\theta\ W_0$$

Using Glaurt (10) equation, the wall jet velocity, thickness and volume flow can be estimated:

$$Q = Q_0\ (Y/Y_0)^{0.946}$$

The increment in jet volume and therefore in air entrainment caused by advance of the jet front on the wall can now be evaluated.

$$E_{wall} = \rho a \int_t^{t+\Delta t} Q \, dt$$

Spray Graphical Presentation A spray global presentation based on the spray geometry has been developed. (Appendix 1).

This presentation as elucidated in Fig (3) take account of wall jets interaction. Once the jets are interacting the air in areas of interaction is shared accordingly between adjacent jets.

Momentum Losses. In the jet interaction model described previously, assumptions leading to evaluation of air entrainment are not strictly valid. This is because momentum is not fully conserved. In order to take account of momentum losses, an entrainment factor EF is defined as:

$$E_{Actual} = EF \cdot E_{Calculated}$$

Intimate Mixing. While the air entrained by the gas jet at any instant quantifies the larger scale mixing of fuel and air in the chamber, intimate mixing is represented by the following expression.

$$dMa/dt = DC\ V_f\ (E-Ma)$$

Integration leads to

$$Ma = E\,[1 - 1/(e^{Lt})]$$

where $L = DC.V_f$

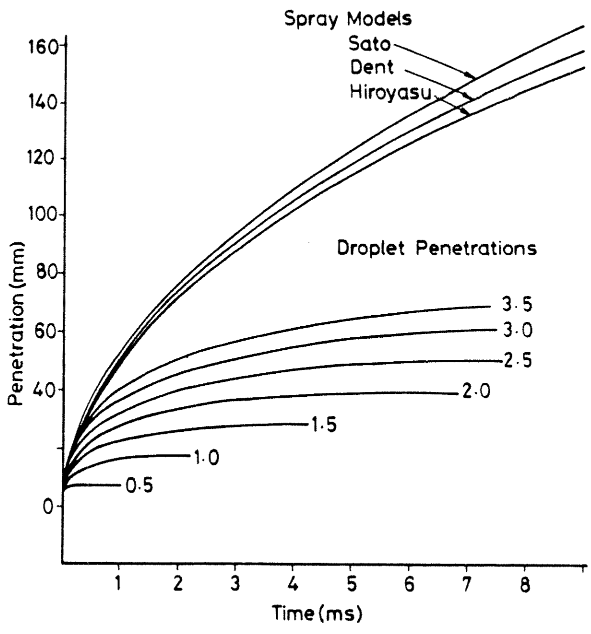


Fig. 4 Jet Penetration: Spray vs. Droplet Model

Hence a value for the heat released can be calculated. Applying the first law of thermodynamics results in a value for the corresponding work done for a given mix of fuel and air.

Heat Transfer. The heat transfer formula used is that formulated by Annand (11) which is based on the actual cylinder/piston surface areas and is calculated step-by-step throughout the processes under consideration.

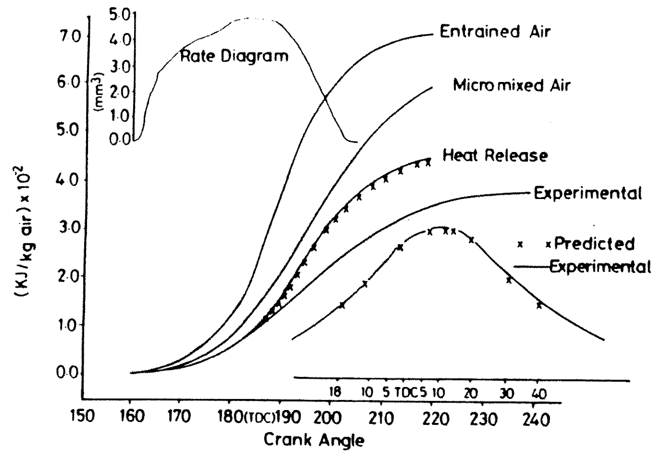


Fig 5 Comparison Between Experimental and Predicted Results for Engine B

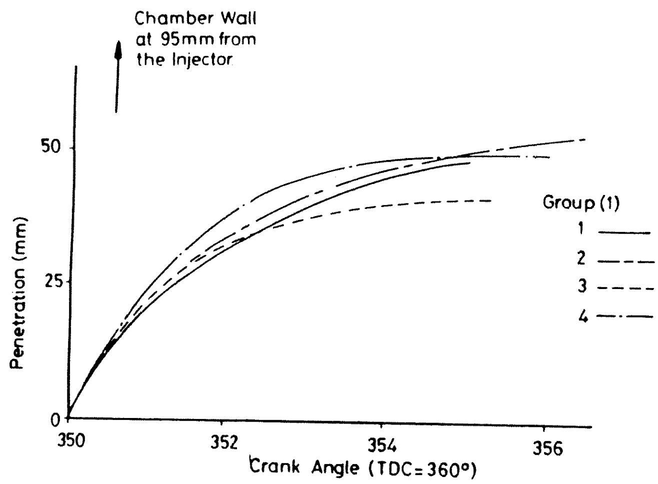


Fig 6 Visible Penetration - Group (1)

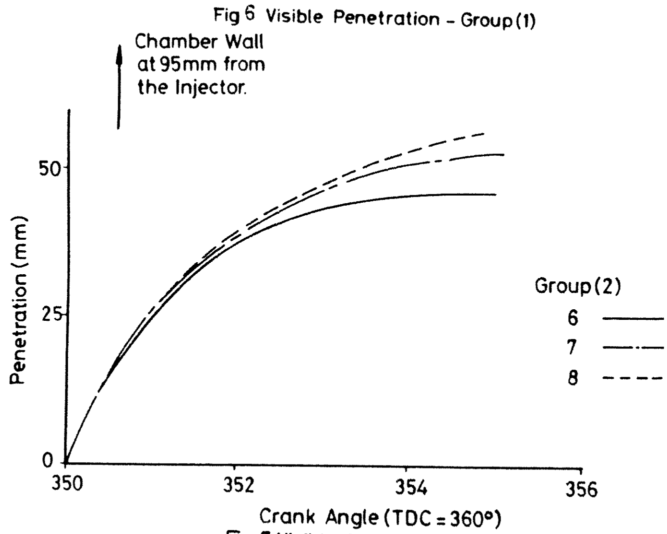


Fig.7 Visible Penetration - Group (2)

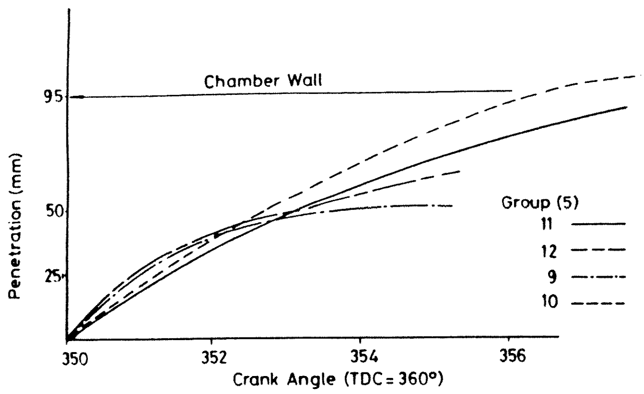


Fig 8 Visible Penetration - Group (3)

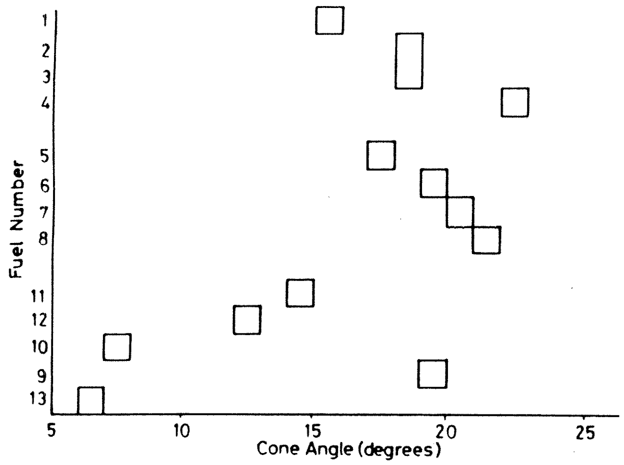


Fig 9 Spray Cone Angle

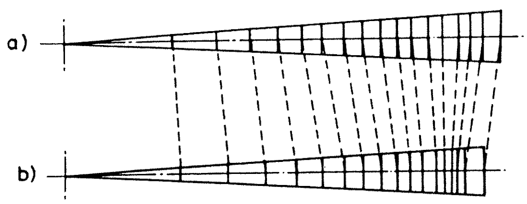


Fig 10 Comparison of Elemental Spray Penetration
a) Mean Injection Pressure
b) Actual Injection Pressure

Expansion (Process 3 - 4). The expansion process is described by a similar relation to that for the compression period.

Blow Down Period (Process 4 - 5). Amongst all the open cycle periods the blow down is considered more significant.

$$T5 = T_4 (p5/P_4)^{\frac{n_b-1}{n_b}}$$

Overall Cycle Parameters. The work done during each period is calculated including those for the Exhaust, Overlap, Suction and Pre-compression processes. The total work done is then evaluated. The other overall parameters are calculated according to (12).

Fuel Injection Program

This program is only used when experimental injection diagrams are available.

An expression is obtained by applying continuity, neglecting transient effects:

Flow through the orifices =
Flow through the annular area =
Flow through the nozzle holes i.e.,

Therefore:

$$Q = K_e \; A_e \; \sqrt{(P_{diff}/RPM)}$$

Heat Release Program

For analysis only. The experimental values of cylinder pressure are used in conjunction with corresponding cylinder volume to calculate both the work done and the internal energy. Utilising the 1st law of thermodynamics heat release can be calculated for each element.

Temperature is calculated as follows:

$$T = \frac{P.V}{M_T.R}$$

This temperature is then used to evaluate the internal energy (U). Knowledge of P, V and U enables the heat release to be found by applying the 1st law of thermodynamics.

Heavy Fuel Consideration

The penetration rate and cone angle expressions already take account of the effect of heavy fuels. To this end, the air entrainment calculations do apply irrespective of the type of fuel used . However, as far as the delay period, and, in turn the initial rate of pressure rise is concerned, droplet consideration needs to be taken into account. The description of the full combustion model primarily based on approach 2 is beyond the scope of this paper. Nevertheless, the presented model in the paper viz Approach 1 has been assigned a sub-model for droplet calculations (5).

Prediction vs Experimental Results

Full description of predicted and experimental results is beyond the scope of this paper. Figs 12a to 12d show the comparison between theoretical and experimental results at two different loads viz 100% and 75% for diesel furl as well as a heavy fuel.

CONCLUSION

The ability of the model to predict engine performance parameters to within the acceptable experimental error range involving some forty cases at different test conditions with four different engines, with standard and high pressure fuel injection equipment must be considered very encouraging and a proof of the model. Furthermore the most recent tests carried out on engine D with diesel and heavy fuels clearly indicates the engine model is capable of predicting the behaviour of the medium size diesel engines running of heavy fuels.

Whilst the model needs further refinements it can now be used for various combustion studies including engine/fuel injection matching for engines running on diesel or heavy fuels.

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NOMENCLATURE

P	Cylinder pressure (suffix denotes piston location)
V	Cylinder volume (suffix denotes piston position)
V_{cl}	Clearance volume
M_T	Trapped mass
M_A	Micromixed air
P_A	Piston area
η_v	Volumetric efficiency
T	Temperature (suffix denotes piston location)
T_{exh}	Exhaust temperature
S_{ff}	Effective stroke
R	Gas constant
ρ_{exh}	Exhaust density (exhaust)
ρ_c	Charge density
ρ_f	Fuel density
ρ_t	as ρ_c
RPM	Engine speed (rev/min)
M_f	Fuel mass
	Mean injection pressure
P_J	Injection Pressure
T_I	Injection period
	Number of holes
C_d	Coefficient of discharge
T_a	Temperature (air zone)
T_p	Temperature (product zone)
E	Entrainment
V_f	Jet tip velocity (w for the wall jet)
DEL	Delay
C_n	Cetane number
N_b	Blow down period index
A_e	Nozzle effective orifice area

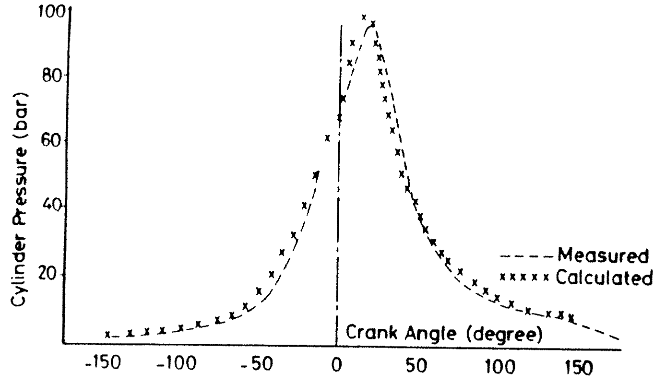


Fig 12a Cylinder Pressure Diesel Fuel Full Load

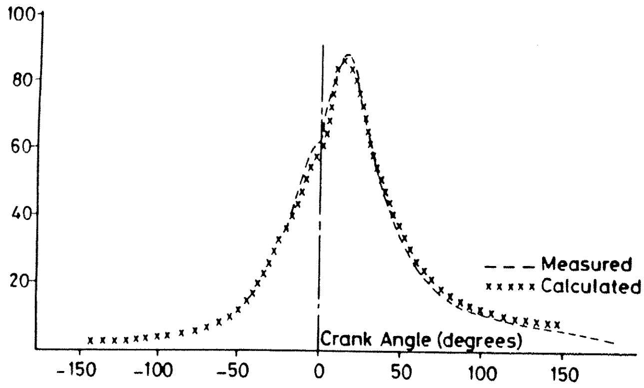


Fig.12b Cylinder Pressure Diesel Fuel 75%

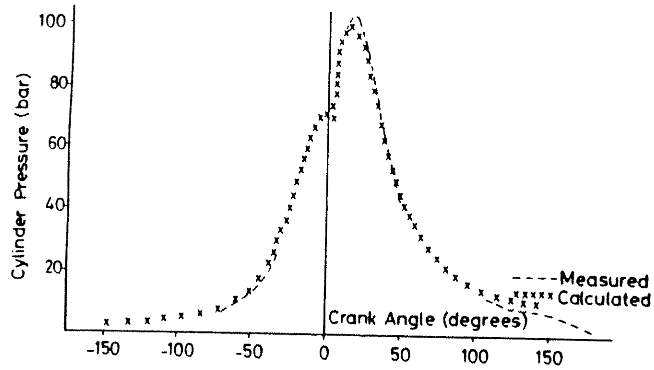


Fig 12c Cylinder Pressure Heavy Fuel Full Load

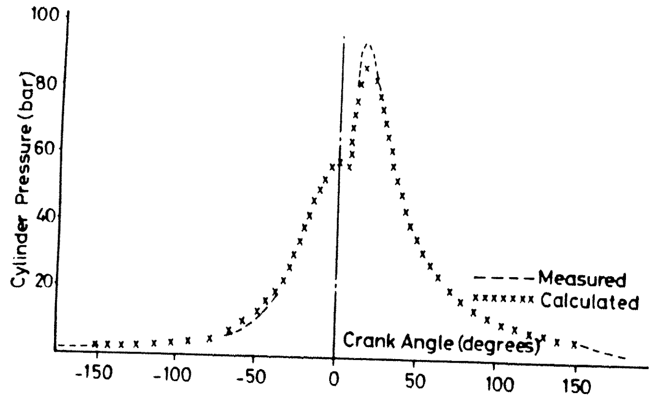
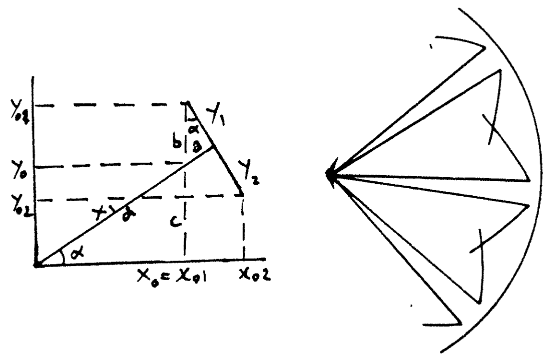


Fig12d Cylinder Pressure Heavy Fuel 75% Load

P_{diff} Total pressure drop accress the nozzle effective area
Q Fuel flow through the nozzle
A Piston area
F_{pen} Penetration
F Maximum penetration
t_T Fuel to vapour transition time
t_{fw} Impingement time
$Y(Y_0)$ Wall jet radius (initial value)
$S(S_0)$ Wall jet thickness (initial value)
$W(W_0)$ Wall jet velocity (initial value)
$Q(Q_0)$ Wall jet volume flow

APPENDIX 1

SPRAY GRAPHICAL PRESENTATION



SPRAY GLOBAL PRESENTATION

yo2 = c+b ; $Xo=(d^2-c^2)^{1/2}$

c=Xo tanα ; a=Y1 tanα

b=Y1/cosα ; d=X1-Y1 tanα

Therefore: yo1=Xo tanα + Y_1/cosα

$Xo1=\sqrt{[(X1 - Y1 \tan\alpha)^2 - Xo \tan\alpha]}$

Yo2=Yo1-(Y1+Y2)/cosα

Xo2=Xo1+(Y1+Y2)sinα

c=d sinα =(x1-y1 tanα) sinα

yo1=(X1-Y1 tanα)-sinα +Y_1/cosα

$Xo1 = \sqrt{\{(X1-Y1 \tan\alpha)^2-[(X1-Y1 \tan\alpha) \sin\alpha]^2\}}$

Where: $X1 = F_{pen}$

$Y1 = Y2 = W_{pen}$

APPENDIX 2

DELAY EXPRESSIONS

For diesel fuel

$$DEL = 15.375 \times 10^{-3} \text{ RPM } C_n^{-a} \cdot P_j^{-0.38} \cdot e^{AE/T} + 39.04 \Delta P^{-0.35} \cdot d_n^{0.4}$$

Where a = 0.7 to 1.0 and AE the activation energy

For heavy fuels

DEL = DL + DC.CCAI

Where DL and DC are constants and their values depend on the engine load.

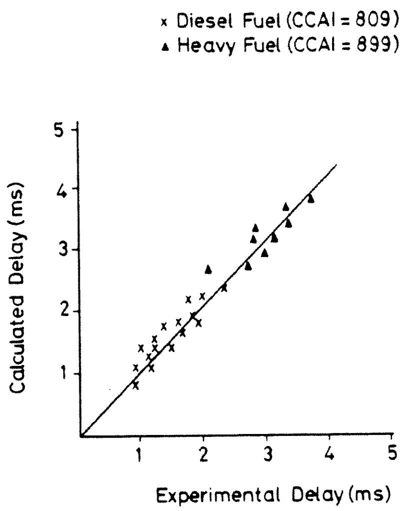


Fig.15 Calculated vs Experimental Ignition Delay

APPENDIX 3

GROUP (1)

Fuels simulating marine diesel oil by varying the front end with constant back end.

Fuel (1) - 15% 'K' Residue, 85% Gas Oil

Fuel (2) - 15% 'K' Residue, 85% Gas Oil

Fuel (3) - 15% 'K' Residue, 85% Gas Oil

Fuel (4) - 15% 'K' Residue, 85% Gas Oil

GROUP (2)

Fuels simulating marine diesel oil by varying the back end and constant front end.

Fuel (5) - 15% 'L' Residue, 85% Gas Oil

Fuel (6) - 15% 'V' Residue, 85% Gas Oil

Fuel (7) - 15% 'I' Residue, 85% Gas Oil

Fuel (8) - 15% 'C' Fuel, 85% Gas Oil

GROUP (3)

Fuels from pre-dominantly single crude sources.

Fuel (9) - 'C' Marine Diesel Oil

Fuel (10) - 'C' Fuel Oil

Fuel (11) - 'NS' Residue

Fuel (12) - 94% 'K' Residue, 6% Gas Oil

Fuel (13) - 100 'G' Residue

APPENDIX 4

ENGINE	A	B	C	D
NO. OF CYLINDERS	1	MULTI	1	MULTI
BORE mm	215	190	114	203
STROKE mm	241	210	139	273
SWEPT VOLUME cc	8833	5954	1433	8836
COMP.RATIO 'NOMINAL'	11	12	11	12.2

Table 1 Engines Geometry

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