PREDICTION OF PHYSICAL DELAY PERIOD IN DIRECT INJECTION DIESEL ENGINE COMBUSTION

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Abstract A semi-empirical mathematical model for predicting the physical part of ignition delay period in the combustion of diesel engines with swirl is developed. This model is based on a single droplet evaporation model. The governing equations, namely, equations of droplet motion, heat and mass transfer were solved simultancously using a Runge-Kutta step by step method. The computation was executed until somewhere in the vapor layer around the liquid droplet a near stochiometric mixture of the fuel vapor and air having at least the self-igniton temperature of the fuel formed. The predicted physical delay time for a particular D.I. diesel engine is in good agreement with engine standard data and data in the literature. Also validity of the model is examined with variation of the combustion chamber and fuel injection system data. From the parametric studies it seems that the physical delay period is particularly effected by fuel initial temperature, injection pressure, swirl level, and ambient temperature. Also from examination of the results an algebric relation for quick calculation of physical delay time is derived.

چکیده جهت پیش بینی زمان مربوط به قسمت فیزیکی تأخیر در اشتعال در احتراق موتورهای دیزلی با پاشش مستقیم و با حرکت هوای گردبادی، یک مدل نیمه تجربی ریاضی ساخته شده است. این مدل براساس مدل «تبخیر قطره سوختی» قرار دارد. معادلات حاکم یعنی معادلات حرکت قطره، انتقال گرما و جرم، بطور همزمان با استفاده از روش عددی رانگ - کوتا حل شده‌اند. محاسبات تا وقتی ادامه می یابند که در لایهٔ اطراف قطره سوختی یک مخلوط نزدیک به استوکیوتریک با دمای اشتعال خودبخودی سوخت تشکیل گردد. زمان تأخیر فیزیکی پیش بینی شده از این مدل با نتایج تجربی استاندارد موتور دیزلی موردنظر و با نتایج موجود در ادبیات فن بخوبی مطابقت دارد. همچنین برای تائید مدل حاضر، مطالعات پارامتریک با استفاده از داده های محفظه احتراق و سیستم پاشش سوخت بعمل آمدند. این مطالعات نشان می دهند که پریود تاخیر فیزیکی تحت تأثیر دمای اولیه سوخت، فشار پاشش سوخت، شدت گردباد و دمای محیط سیلندر قرار می گیرد. همچنین از بررسی تحلیلی نتایج بدست آمده، رابطه جبری ساده‌ای برای محاسبه سریع تأخیر فیزیکی بدست آمده است.

INTRODUCTION

Today the main target in the field of diesel engineering research is to improve fuel economy and reduce emission and noise levels from high-speed diesel engines. These performance qualities are mainly governed and controlled by the nature of the combustion process which takes place in these types of engines. Now it is well known that the combustion process in diesel engines occurs in nearly four distint stages [1], that is,a) ignition delay period,b)rapid combustion period or premixed type combustion,c) moderate rate combustion or diffusion type combustion and d) slow rate combustion period or tail of combustion. From engine performance, emission, and heat release studies, it is understood that, ignition delay period plays an important role in the combustion process. This is becuse its length really dictates burning rates and proportion of mass burnt in the different stages of combustion process. Hence engine performance qualities such as starting, roughness, noise level, and emission will be affected by the ignition delay time period. So it is worthwhile to investigate the ignition delay period in more detail.

When liquid fuel is injected from an injector into the cylinder of diesel engines, it leaves the injector tip initially as liquid ligaments and after a short period of time due to shear forces, it breaks up into a fuel spray consisting of many droplets of various sizes. Because of the lower surface/ volume ratio of the ligaments, vaporization of liquid fuel during this time period is small and neglected. Some researchers

[1, 2] consider the jet break up period as the first part of ignition delay period.

When droplets are formed, because of their large surface to volume ratios, vaporization of liquid droplets begins. Before vaporization starts, droplets must receive heat from the surroundings until their outer surface temperature reaches the fuel boiling point. From now on vapor formation starts in the outer layer of the droplet surface and therefore a thin layer of fuel vapor mixed with air is created on the droplet evaporating surface.

As vapor forms it receieves heat from hot air and becomes superheated until somewhere in the mixed layer a near stoichiometric mixture of the fuel vapor and air having at least the self ignition temperature (SIT) of the fuel is formed. The time elapsed between formation of droplets and preparation of combustible mixture of fuel vapor and air is defined as the physical part of ignition delay. It seems that this situation is usually formed at the spray edge when little or no interaction between droplets is taking place.

After the formation of a combustible mixture somewhere in the thin layer, the diffusion process takes place due to a concentration gradient between the thin layer outer surface and the surrounding heated air. And perhaps within the prepared mixture of thin layer, a flame nuclea created. The time period between the end of physical delay and creation of flame is defined as chemical delay. As described above ignition delay time in diesel engine combustion consists of nearly three periods as follows:

1) Jet break up period
2) Physical delay period
3) Chemical delay period

Most workers have concentrated to a large extent on considering liquid jet as gaseous (or vapor) in form and hence existance of droplets were neglected [3-6]. Therefore two important parts of ignition delay, that is, jet break up and physical delay periods were ignored and so far almost all the chemical part of ignition delay was considered as a whole ignition delay period [3,5,6]. This is because no distinct and observable event occurs at the end of physical delay period. Only a few researchers [2, 7, 8, 9] in this field have investigated the ignition delay with its three components. Their mathematical models are based on single droplet evaporation model for physical delay and Arrhenius type equation for chemical delay calculation. Their models are not suitable for high-speed diesel engines, since they have not considered moving-droplet evaporation process within swirling air inside the combustion chamber.

In this work a simple equation is used for calculation of jet break up period and for physical delay period a semi-empirical single droplet model is used which has been modified to take droplet motion relative to swirling air into account. In this model, physical delay period was terminated when within the assumed thin layer around the droplet a combustible mixture was formed.

MODEL DESCRIPTION

From the above discussion it can be concluded that during the first parts of ignition delay period a set of physical phenomena occurs as follows:

-Liquid fuel injection as a liquid jet
-Liquid jet break up and droplet formation
-Vaporization of droplets due to heat, mass, and momentum transfer with surrounding air
-Formation of combustible mixture within the thin layer around the droplet surface

Present physical delay model is based on the droplet evaporation theory and it must be handled mathematically in order to construct the prediction model. As mentioned before jet break up period is considered as the first part of ignition delay time and is calculated from the following equation [2]:

$$t_b = \frac{\alpha . \rho_L . d}{c\sqrt{2\rho_a \Delta P}} \qquad (1)$$

where, α = 15. 8 and c = 0.8

For vaporization of the moving droplet, equations of mass, heat and momentum transfer

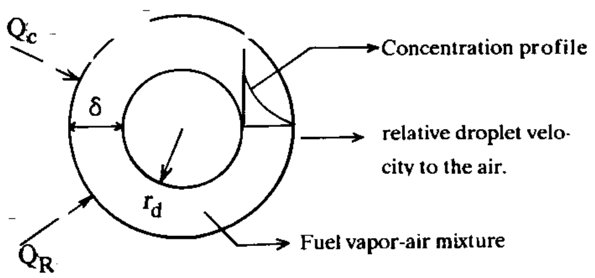

Figure 1. Single moving- droplet evaporation model

are considered. Figure 1 shows the single moving droplet evaporation model.

A semi-empirical equation is used here to determine the mass transfer rate from the surface of the droplet. This equation arises from consideration of molecular mass-diffusion rates in a layer of fuel-vapor assumed to exist on the surface of the droplet. The mass equation is empirically modified to take into account the effect of convection due to the motion of the droplet relative to the swirling air. Heat transfer rate, from surroundings to the droplet included the use of a semi-empirical equation taking into account convective and radiative heat transfer effects. In the previous work [1] radiation heat transfer was not considered. The droplet velocity was determined equating the rate of change of momentum of the droplet to the drag force upon it. The equations of mass and heat transfer are first-order but the momentum equation is second-order, which was transfered to two first order eqations.

For time history of moving- droplet the above four differential equations were solved as

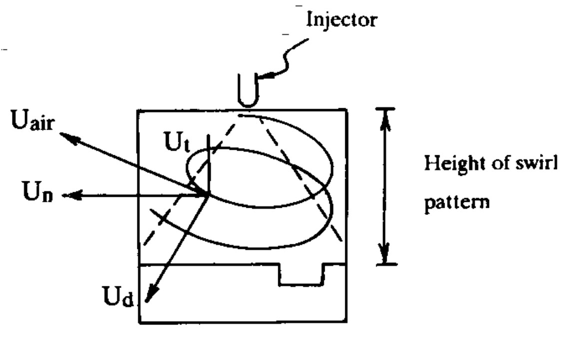

Figure 2. Schematic view of air swirl pattern

functions of time. Also a simple pattern for calculating air swirl velocity was assumed. Figure 2 illustrates the schematic view of the air swirl pattern used.

Complete analytical method for calculation of air swirl velocity is given in Appendix A. From this type of calculations effective air swirl velocity is found as:

$$(U_a)_{eff.} = 0.7\left(\frac{b}{2}\ \omega\right) \tag{2}$$

For calculation of combustible mixture, air/ fuel ratio around the moving-droplet the following relation was considered:

$$AFR = \frac{4\pi/3(R_0^3 - R_1^3)\rho_a}{\frac{dm_{ev}}{dt}.\Delta t} \tag{3}$$

Assumptions Used

To simplify the calculation procedure the following assumptions were made:

1) The interactions of droplets are ignored. This is acceptable in the region at the edge of the spray.

2) The spherical symmetry for droplet and the mixture layer is assumed until the end of the physical delay period.

3) Temperature and density gradients inside the droplet are ignored.

4) Temperature distribution is uniform throughout the combustion chamber.

NUMERICAL SOLUTION

Four ordinary differential equations which were pointed out above were solved using a step- by-step fourth-order Runge-Kutta method. After selection of a suitable time step and initial values for variables, thermodynamical properties of liquid fuel/ air, and mixture are calculated. Hence by the subroutine FCT the right- hand sides of the four equations are determined. For calculation of droplet relative velocity subroutine SWIRL is utilized. Subroutine AFR was written to calculate air/ fuel ratio within the thin layer around the moving droplet.

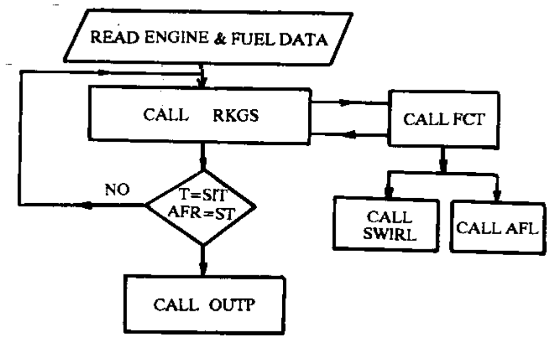

Figure 3. Flowchart of model

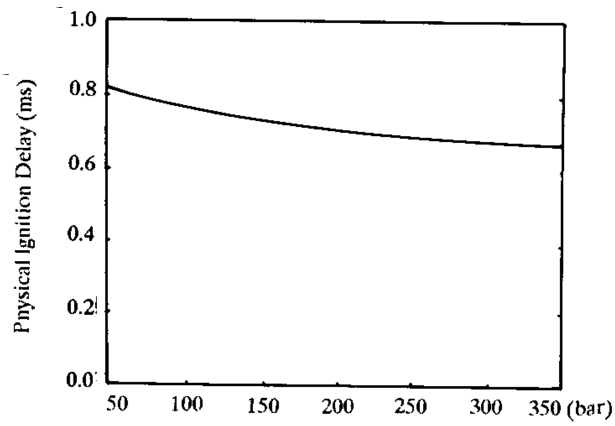

Figure 4. Effect of injection pressure on physical ignition delay

When conditions for end of physical delay were satisfied, the calculation terminated. Figure 3 shows the flow-chart for calculation sequence of the main program with its relevant subroutines.

STANDARD RESULTS

Table 1 shows the engine standard data with calculated physical delay time for standard conditions.

Table 1. Engine Data and Results

Parameters	Data
Injection commencement	21^0CA BTDC
Compression ratio	16.8
Engine speed	2300 rpm
Injection pressure	200 bars
Swirl ratio	7
Initial fuel temperature	400 K
Ambient pressure	101325 Pa
Ambient temperature	298 K
Rated power	130 kw
Ignition Delay (Manufacturer)	1.1ms(15CA)
Calcutated jet break up period	0.1 ms
Calculated physical delay period	0.71 ms

Table 1 gives the calculated time for two initial parts of ignition delay period for this engine. It can be seen that most of ignition delay period has been allocated to physical delay (i.e. 64.5%)

PARAMETRIC STUDIES

In order to examine the validity of the present model, the effect of variation of the important parameters of the engine combustion chamber and fuel injection system on the physical delay is also investigated. Calculations show that the effect of following parameters are important:

- Injection pressure
- Fuel initial temperature
- Ambient temperature
- Air swirl ratio
- Air/ Fuel ratio
- Engine speed

Injection Pressure

Figure 4 shows the effect of variation of the injection pressure on the physical delay. As shown, by increasing injection pressure, physical delay decreased. This is because as injection pressure increases the mean droplet diameter decreases and also droplet initial velocity increases. Both of these effects will speed up the evaporation process and consequently physical delay will be decreased. from Figure 4 which is based on computer results, applying least squares method, the following relation can be obtained:

$$PHDP_{inj} = 0.8262 \ \exp (P_{ing}) \tag{4}$$

Fuel Initial Temperature

Figure 5 shows the effect of variation of the fuel initial temperature on the physical delay.

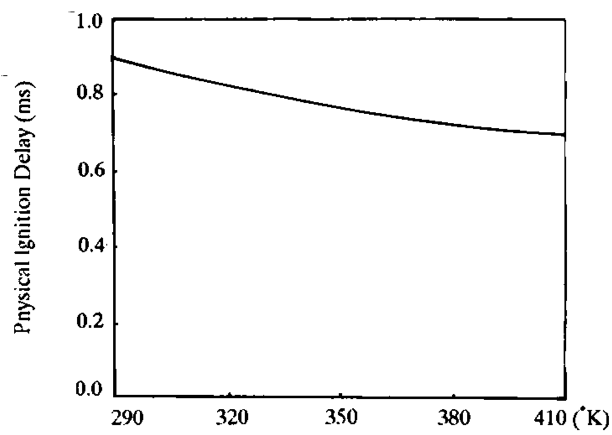

Figure 5. Effect of fuel initial temperature on physical ignition delay

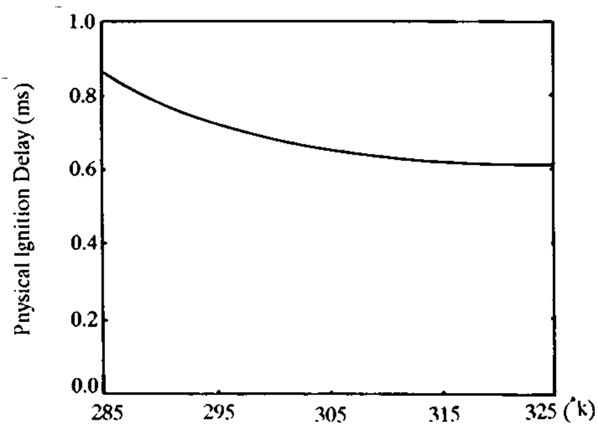

Figure 6. Effect of ambient temeperature on physical ignition delay

Increasing this parameter also decreases the physical delay time. This parameter has significant influence on the quality of droplet formation,as it affects the fuel properties, such as density, viscosity, surface tension, specific heat, diffusivity, and latent heat of evaporation. All these properties are the functions of liquid fuel temperature. By increasing this parameter the rates of formation of droplets and heat transfer improved, which results in a shorter physical delay. Again by using least squares method we have:

$$PHD_{T_d} = 1.6344 \exp(-0.0021\, T_d) \qquad (5)$$

Ambient Air Temperature

Figure 6 shows the effect of variation of ambient air temperature on the physical delay period. As it is shown, by increasing air temperature the physical delay reduces. From engine cycle calculation it is well known that increasing inlet air temperature will increase the temperature of the whole cycle and hence temperature at the commencement of injection. This will cause a steep temperature gradient between droplet and surrounding heated air, which enhances the process of heat transfer as well as vaporization, so this will shorten the physical delay. Here also we have:

$$PHD_{T_a} = 5.1376 \exp(-0.00643\, T_a) \qquad (6)$$

Air/ Fuel Ratio

Figure 7 illustrates the effect of variation of air/ fuel ratio within the mixed layer around the droplet. As it shows, increasing the air/fuel ratio causes the physical delay to become longer. This means that by varying mixture strength the time required to form stoichiometric mixture around the droplet becomes longer. For this variation we have:

$$PHD_{AFR} = 0.01307\, AFR + 0.5201 \qquad (7)$$

Swirl Ratio

Figure 8 shows the effect of variation of swirl ratio ($SR = \omega / (2\pi N)$) on the physical delay.

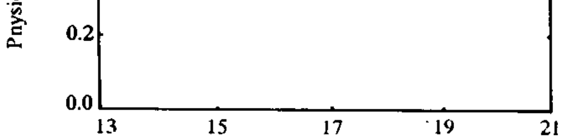

Figure 7. Effect of air/ fuel ratio on physical ignition delay

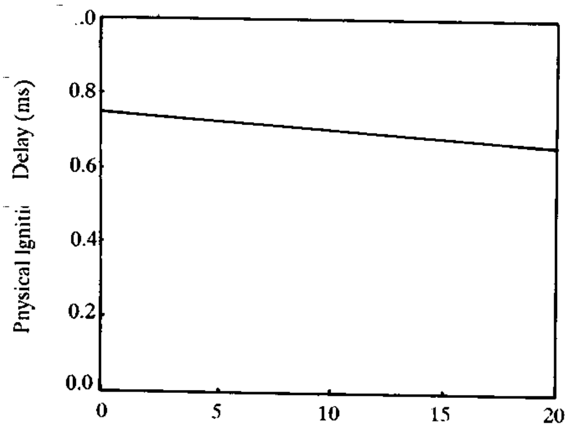

Figure 8. Effect of swirl ratio on physical ignition delay

Higher swirl ratios enhance heat transfer and evaporation processes, so physical delay time will be decreased. In other words creation of turbulence within the combustion chamber improves the heat transfer coefficient and mixing process. Applying the least square method to the computer results yields the following equation:

$$PHD_{SR} = -0.0075\ SR + 0.77 \qquad (8)$$

Engine Speed

Figure 9 shows the physical delay as a function of engine speed. As shown in this figure the effect of this parameter on physical delay at lower and higher speeds is considerable but at moderate engine speeds physical delay is nearly constant, From the results it can be derived that:

$$PHD_N = 0.6645 + 1.6028 \times 10^{-4} N - 6.143 \times 10^{-8} N^2 \qquad (9)$$

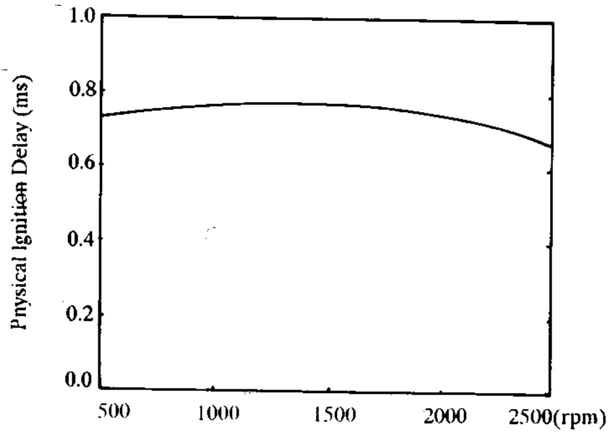

Figure 9. Effect of engine speed on physical ignition dealy

By combining the above correlations, the average physical delay period may be obtained from geometric mean as follows:

$$PHD_{av} = \sqrt[6]{PHD_{P_{inj}} . PHD_{SR} . PHD_{Td} . PHD_{AFR} . PHD_{Ta} . PHD_N} \qquad (10)$$

It may be noted that with standard engine data, average physical delay time obtained from the above relation is equal to the one obtained from the present mathematical model.

As mentioned earlier, due to the uncertainity of end conditions of the physical delay period, accurate measurement of this time period is not easily possible. However, for verification purposes of the model, results of parametric studies qualitatively are in good agreement with the results of other investigators (e. g. reference 8) which are obtained theoretically, despite the difference in the location of the fuel vapor formation.

CONCLUSION

On the basis of the previous discussions the following results can be deduced:

1. The present model can be applied with sufficient accuracy for predicting the physical delay time in D. I. diesel engines.
2. It seems that the main part of ignition delay time is taken by physical delay period.
3. From parametric studies it can be seen that fuel initial temperature and inlet air temperature have great influence on physical delay period.
4. From parametric studies a simple algebraic correlation for easy calculation of the physical part of ignition delay is obtained.

NOMENCLATURE

A	Nozzle type constant
AFR	Air/ fuel ratio
B	Amount of fuel delivered, (m^3/stroke) (eq A-6)
b	Cylinder bore diameter, (m)
C	Drag coefficient, discharge coeff.
D	Diffusion coefficient,(m^2/sec)

d	Nozzle diameter, (m)
F	Drag force, (N)
M	Molecular weight
m	Mass, (kg)
N	Engine speed,(rpm)
Nu	Nusselt number
P	Pressure, (Pa)
Pb	Piston bowl dia.,(m)
PHD	Physical delay time, (sec)
Q	Heat transfer rate, (watt)
R	Universal gas constant,(J/kg^0K)
r	Radius of droplet, (m)
S	Distance of piston crown from cylinder head at any crank angle position (m)
SMD	Sautor mean diameter, (μm)
SR	Swirl ratio
T	Temperature,(K)
t	Time,(sec)
U	Velocity,(m/s)
V	Combustion chamber volume, (m^3)
x,y,z	Cartesian coordinates

GREEK SYMBOLS

α	Nozzle type constant
β	Angle of inclination of helix to generator, (rad)
θ	Patameter (angle of revolution), (rad)
δ	Thin layer thickness (vapor + air), (m)
Δ	Difference
π	Pi (const, 3.1415)
ρ	Density, (kg/m)
ω	Angular speed, (rad/ sec)

SUBSCRIPTS

a	Air
av	Average
c	Convection
d	Droplet
do	Droplet initial
ev	Evaporated
f	Fuel
fl	Fuel vapor
	Inner radius
inj	Injection
L	Latent, Liquid
n	Normal
o	Outer radius
R	Radiation
rel	Relative
t	Tangent
x,y,z	Coordinate directions

Appendix A

Equations of Droplet Motion and Evaporation

$$\frac{d^2x_d}{dt^2} = -F\frac{dx_d}{dt}$$

$$U_d = \frac{dx_d}{dt} \qquad \text{(A-2)}$$

$$\frac{dm_{ev}}{dt} = \frac{-2\pi D.P_{fL}.NU_m.\alpha.MWF}{R.T_{av}} r_d \qquad \text{(A-3)}$$

$$= Q_c + Q_R - Q_L$$

Where, $F = (1.5708)\rho_a.r_d^2.U_{rel}.C_d/m_d$

$$U_{rel} = (U_d^2 + U_a^2)^{0.5}$$

$$m_d = \frac{4}{3}\pi r_d^3.\rho_L$$

$$T_{av} = \frac{T_a + T_d}{2}$$

$$NU_m = 2 + 0.6\, R_e^{\frac{1}{2}}.S_c^{\frac{1}{3}}$$

Droplet initial velocity:

$$U_{d_o} = c\sqrt{\frac{2\Delta P}{\rho_L}}$$

Sautor Mean Diameter [2]:

$$SMD = A(\Delta P)^{-0.135}.(\rho_a)^{+0.121}.(B)^{0.131}$$

$$A = 2.33 \times 10^{-3} \text{ for hole nozzle (const.)} \qquad \text{(A-6)}$$

Appendix B

Calculation of air swirl velocity. Figure 3 shows the swirl pattern used in this model. The parametric equations of swirl helix are,

$$x=\frac{b}{2}\cos\theta$$

$$y=\frac{b}{2}\sin\theta \qquad \text{(A-7)}$$

$$z=\frac{b}{2}\theta \operatorname{cotg}\beta$$

By taking derivation with respect to time we have:

$$U_x=\frac{\dot{b}}{2}\cos\theta - = \frac{b}{2}\dot{\theta}\sin\theta$$

$$U_y=\frac{\dot{b}}{2}\sin\theta + \frac{b}{2}\dot{\theta}\cos\theta \qquad \text{(A-8)}$$

$$U_z=\operatorname{cotg}\beta\left(\frac{\dot{b}}{2}\theta+\frac{b}{2}\dot{\theta}\right)$$

$$|U_a|=\sqrt{U_x^2+U_y^2+U_z^2} \qquad \text{(A-9)}$$

Since during physical delay period the height of combustion chamber is too small, the U_z component of air velocity could be omitted. Thus we have,

$$|U_a|=\sqrt{U_x^2+U_y^2}$$

After substitution from eq (A-8) we have,

$$|U_a|=\sqrt{\left(\frac{\dot{b}}{2}\right)^2+\left(\frac{b}{2}\dot{\theta}\right)^2} \qquad \text{(A-10)}$$

Since the first term in eq. (A-9) in comparision with the second term is too small, it can be neglected.

From Ref. [10, 11] effective velocity of air swirl can be taken as:

$$|U_a|_{eff}=0.7\left(\frac{b}{2}\omega\right) \qquad \text{(A-11)}$$

$\omega = \dot{\theta}$ and is adopted form Ref. [12] as:

$$\omega=\frac{b^2}{2}\omega_o/ \quad \frac{\pi(b/2)^4.S(\theta)/V+(Pb)2/4}{\pi(b/2)^2.S(\theta)/V+1} \qquad \text{(A-12)}$$

Appendix C

Table 2. Engine Specifications

Parameters	Data Values
No of cylinders	6
Bore diameter	115 mm
Piston stroke	140 mm
Compression ratio	16.8
Rated speed	2300 rpm
Relative fuel/ air ratio	0.55
Con.rod length	234 mm
Start of injection	21°BTDC
Rated power	130 kw
Piston bowl	height=22mm
(Cylindrical)	diameter=65mm

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