A SIMULATION MODEL FOR DIESEL ENGINE CYCLE ANALYSIS AND COMBUSTION

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ABSTRACT

An analytical model for the description and prediction of diesel engine cycle analysis over a wide range of operating conditions is presented. Such model of cycle analysis and combustion is very useful for the design and development of internal combustion engines. The model takes into account the piston displacement , gas flow through valves, conservation laws of mass and energy, heat transfer correlations in engine cylinder and the calculation of the fuel burning rate based on the Whitehouse-Way`s method. For accurate computations; a resolution (from 2 to 5) degrees crankangles was convenient for gas exchange processes, while a resolution less than one degree crankangle was needed for the rapid stages of the combustion process. Results of the different processes of engine cycle analysis including gas exchange processes, fuel preparation and burning rates, cylinder gas pressure and temperature, and the heat release pattern are presented.

Keywords: Simulation, Diesel Engine, Cycle Analysis, Combustion.

INTRODUCTION

Simulation models for diesel engine cycle analysis and combustion have become an established part for engine design, development and performance prediction, [1,2]. The earlier attempts by Lyn and Austen, [3-5] aimed at basic understanding of the nature of combustion in diesel engines. They investigated the relation between the rate of fuel injection and the fuel burning rate. They also studied the effect of heat release on the cylinder gas pressure pattern and cycle thermal efficiency.

The simulation model developed by Watson *et al.* [6] was based on algebraic expressions describing the fuel burning rate as a function of the dominant controlling parameters, such as ignition delay and equivalence ratio. Relating the burning rate to these parameters enabled the heat release rate at one engine running condition to be linked to that of another engine. Miyamoto *et al.* [7] developed a more detailed model using the Wiebe`s functions. Their investigation described, with a simple empirical model, the rate of combustion in diesel engine for a wide range of engine operating conditions and different types of engines.

Whitehouse and Way [8-10]. developed a method for calculating the heat release rate suitable for performance and simulation calculations. The equations used were based on a single-zone model for conditions in the cylinder. Their correlation depended on a simple model for the rate of mixing between fuel and air. Besides, the correlation involved the chemical kinetics of the burning process in order to predict the initial burning rate. Their experimental investigations covered a wide range of direct-injection diesel engines; two-stroke and four-stroke engines, small and medium size engines, over ranges of power, speed and air supply conditions.

Fluid dynamics of air flow through intake valve and in-cylinder flow including air swirl and squish flow patterns were thoroughly investigated, [11-13].

Multi-zone models of combustion were developed depending mainly on the submodels of fuel atomization, penetration, evaporation and air mixing, [14-17]. Complete three-dimensional models including submodels for intake flow, air motion, fuel injection and atomization, heat transfer, combustion kinetics, etc., were constructed for diesel engine simulation modelling , [18-20].

The present study aims at constructing a thermodynamic model for cycle analysis and combustion in direct-injection diesel engines. The simulation model includes gas exchange processes, conservation laws of mass and energy, piston displacement and velocity, actual valve timing and a submodel for fuel burning rate based on the Whitehouse-Way`s method. Results of the model show all engine processes including intake, exhaust, overlap period, variation in heat transfer coefficient, heat release pattern and pressure history diagrams. The model covers wide ranges of engine operating parameters. The computer main program is adapted to receive submodels of nitrogen oxides and smoke formation, in the future, to calculate engine emissions. Samples of the simulation results for different cycle processes and engine operating parameters are presented.

PISTON DISPLACEMENT

At any crankangle (θ) measured from the top dead centre, the corresponding piston displacement, charge volume inside the cylinder and the instantaneous piston speed can be calculated from the following relations, respectively [11, 21]:

$$x = (L_C + r) - \left[r \cos\theta + (L_C^2 - r^2 \sin^2 \theta)^{1/2} \right] \quad (1)$$

$$V = V_c + X.\ A_P \quad (2)$$

$$v_p = \left(\frac{2\pi N}{60}\right) r \sin\theta \left[1 + \frac{r \cos\theta}{(L_C^2 - r^2 \sin^2 \theta)^{1/2}} \right] \quad (3)$$

GAS EXCHANGE PROCESSES

In a gas exchange process either the intake valve, the exhaust valve or both valves are opened. Starting from the top dead center, the initial assumption of the composition of gases in the clearance volume (the residual gases) can be calculated assuming complete combustion as follows:

$$C_y H_z + \lambda \left(y + \frac{z}{4}\right)(O_2 + 3.76\, N_2) = y\, CO_2 + \frac{z}{2} H_2O + (\lambda - 1)\ y + \frac{z}{4}\ O_2 + 3.76\,\lambda \left(y + \frac{z}{4}\right) N_2 \quad (4)$$

In order to start computations, suitable values of cylinder pressure and temperature are considered as initial assumptions. Then the mass fraction and the gas constant for each constituent of the cylinder charge can be calculated. The intake process is divided into small increments, each of which occupies ($d\theta$) degrees crankagles. The time interval, for a given engine speed (N), can be written as follows:

$$dt = \frac{d\theta}{6N} \quad (5)$$

Applying the law of conservation of energy to the gas exchange process, [22, 23] during the time interval (dt), we get:

$$dQ + dm_i\, (\bar{c}p_i\ T_i) - dm_e\ (\bar{c}p_e\ T_e) = p\, dV + dU \quad (6)$$

The first term, (dQ), represents the heat transfer from outside boundaries to the cylinder charge. The term $(\bar{c}_{pi} T_i)$ is the specific enthalpy of the incoming air in the intake manifold, while the term $(\bar{c}_{pe} T_e)$ is the specific enthalpy of the exit gases leaving the cylinder. The terms, (dm_i dm_e) refer to the mass transfer through the intake and the exhaust valves respectively.

The mass transfer through a valve related to the upstream stagnation pressure (p_o)and stagnation temperature (T_o), the static pressure downstream the flow restriction (p) and a reference area (A_R) during the time interval (dt) is given, as follows [11,21];

$$\frac{dm}{dt} = \frac{C_d A_R P_o}{(R T_o)^{1/2}} (p/p_o)^{\frac{1}{\gamma}} \left[\frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{p}{p_o}\right)^{\frac{\gamma - 1}{\gamma}} \right] \right]^{1/2} \quad (7)$$

When the flow is choked, i.e. $(p/p_o) \leq [2/(\gamma+1)]^{\gamma/(\gamma-1)}$, the appropriate equation is given by [21]:

$$\frac{dm}{dt} = \frac{C_d A_R P_o}{(R T_o)^{1/2}} \gamma^{1/2} \left(\frac{2}{\gamma + 1}\right)^{(\gamma + 1)/2(\gamma - 1)} \quad (8)$$

For the flow into the cylinder through the intake valve, (P_o) is the intake manifold pressure and (P) is the cylinder pressure. On the other hand, for the flow through the exhaust valve, (P_o) is the cylinder pressure and (P) is the exhaust manifold pressure. The most convenient reference is the valve curtain area (A_c), given by [21]:

$$A_c = \pi D_v L_v \quad (9)$$

The charge mass at the start of the time interval is denoted by (m_1) and the mass at the end of the interval (m_2)is calculated from the law of conservation of mass as follows:

$$m_2 = m_1 + dm_i - dm_e \quad (10)$$

The cylinder charge is treated as an ideal gas mixture having an average gas constant and total internal energy as follows:

$$R = \frac{1}{m} \sum_{i=1}^{i=n} m_i R_i, \quad (11)$$

$$U(T) = \sum_{i=1}^{i=n} m_i \bar{c}v_i (T).\, T \quad (12)$$

The Equations used to calculate the thermodynamic properties of gases in terms of temperature are given in References 23 to 25.

The first model for the engine heat transfer was the Nusselt`s model [27, 28]. The heat transfer coefficient (h) is expressed as the sum of the convective coefficient(h_c)and the equivalent radiative component (h_r) in the following expression :

$$h = h_c + h_r \quad (13)$$

$$h_c = 5.41 \times 10^{-3} (1 + 1.24 \bar{v}_p)(p^2 T)^{1/3} \ \mathrm{kW/m^2.K} \quad (14)$$

$$h_r = \frac{4.21 \times 10^{-4}}{(1/\varepsilon_g + 1/\varepsilon_w - 1)} \left[\frac{(T/100)^4 - (T_w/100)^4}{(T - T_w)} \right] \mathrm{kW/m^2.K} \quad (15)$$

where $\bar{v}_p$ is the mean piston speed (m/s), (p) is the cylinder pressure (MPa) and (ε_g, ε_w) are emissivities of gas and wall respectively.

COMPRESSION PROCESS

The actual compression process starts after the inlet valve closure. The non-flow energy equation and the equation of state applied to the trapped cylinder charge can be written as follows [22, 23].

$$dQ = pdV + dU \quad (16)$$

$$pV = mRT \quad (17)$$

The incremental work and the change in internal energy during an incremental step from point (1) to point (2) are given by:

$$p\, dV = \left(\frac{P_2 + P_1}{2}\right)(V_2 - V_1) \quad (18)$$

$$dU = U(T_2) - U(T_1) \quad (19)$$

The cylinder gases are treated as an ideal gas mixture as given by Equations 11 and 12.

COMBUSTION AND EXPANSION

The ignition delay depends on the values of cylinder pressure and temperature during the ignition delay period. Henein and Bolt [29] and Craven *et al.* [30] reported that the ignition delay in diesel engines can be calculated using the Wolfer Equation 31 as follows:

$$\tau = 0.44 p^{-1.19} \exp(4650/T) \qquad (20)$$

In this equation the pressure p is in (atm), the temperature T in (K) and the ignition delay time τ in (ms). The ignition delay period from the start of injection (θ_1) to the start of combustion (θ_2) can be calculated by means of step by step integration as follows [30]:

$$\frac{1000}{6N} \int_{\theta_1}^{\theta_2} \frac{p^{1.19}\, d\theta}{0.44 \exp(4650 / T)} = 1 \qquad (21)$$

In order to study the fuel burning rate, it is important to calculate the amount of fuel injected during the time interval (dt) and the rate fuel injection as follows [32]:

$$dm_{inj} = m_{inj} \,.dt \qquad (22)$$

$$m_{inj} = (\pi / 4)\, D^2_{noz} \,.\, C_d \,(2\, \Delta P_{noz} \,.\, \rho_f)^{1/2} \qquad (23)$$

The nozzle coefficient of discharge can be experimentally estimated according to the nozzle hole configuration and is set to the value 0.9 as given in Reference 32. The differential pressure (Δp_{noz}) is the difference between the nozzle opening pressure and the cylinder pressure.

The non-flow energy equation and the law of conservation of mass applied for the time interval (dt) is given by References 10 and 22 as follows :

$$dm_b .\, CV + dQ = pdv ++dU \qquad (24)$$

$$m_2 = m_1 + dm_b \qquad (25)$$

The quantity of fuel burnt, (dm_b), can be calculated according to the Whitehouse-Way's method. They presented a simple model for calculating the fuel burning rate in diesel engines. The model takes into account both the premixed and the diffusive phases of combustion as follows [9, 10]:

1. When the fuel is injected into the cylinder, it must be atomized, evaporated and mixed with sufficient air for burning. This process is referred to as the preparation rate. For a four-stroke diesel engine the fuel preparation rate is given as follows:

$$\frac{dm_p}{d\theta} = c_1\, m_{inj}^{1/3}\, m_u^{2/3}\, p_{02}^{0.4} \qquad (26)$$

The constant (c_1) in Equation 26 has the value (0.014) for a direct-injection diesel engine. The cumulative mass of fuel injected (m_{inj}) is calculated from the start of fuel injection up to this step of calculations. The cumulative mass of unprepared fuel (m_u) is the difference between the two cumulative amounts (m_{inj}-m_p) up to this point of calculations. The partial pressure of oxygen in the cylinder charge (P_{02}) is in (bar).

2. For the first phase of heat release, i.e. the premixed mode of combustion, there will be enough fuel prepared during the ignition delay period. Therefore the actual rate of fuel burning will be determined according to the reaction rate ($dm_r/d\theta$) as follows, [10]:

$$\frac{dm_b}{d\theta} = \frac{dm_r}{d\theta}, \qquad (27)$$

$$\frac{dm_r}{d\theta} = \frac{c_2 P_{02}}{N\sqrt{T}} \exp(-act / T) \int \left(\frac{dm_p}{d\theta} - \frac{dm_b}{d\theta}\right) d\theta \qquad (28)$$

The integral term,

$$\int [(dm_p / d\theta) - (dm_b / d\theta)]\, d\theta \;,$$

is equal to the quantity of fuel in the cylinder that has been prepared but not yet burnt. The constant of the Arrhenius exponent termed (act) takes the value of (15000 K). The constant (c_2)

in Equation 28 takes the value of [10.5 x 10^{10} $K^{1/2}$ / bar . s] for a direct-injection diesel engine, [9, 10].

3. For the second phase of heat release, i.e. the diffusive mode of combustion, the cumulative amount of prepared fuel will be totally consumed, i.e. at the start of this stage the cumulative amount of reacted fuel will be equal to the cumulative amount of prepared fuel (m_r=m_p). Therefore the fuel burning rate in the diffusive phase of combustion will be equal to the rate of fuel preparation given by Equation 26 as follows:

$$\frac{dm_b}{d\theta} = \frac{dm_p}{d\theta} \qquad (29)$$

4. For each step of calculations, if the cumulative reacted fuel is less than the cumulative prepared fuel, then the rate of fuel burning will be calculated by Equations 27 and 28. Otherwise, if the cumulative amount of reacted fuel consumes all the amount of the prepared fuel, then the rate of fuel burning will follow the diffusive combustion mode and is calculated by Equations 26 and 29.
5. The diffusive combustion continues during the expansion process with small rate of combustion until all the cumulative fuel injected (m_{inj}) is consumed, i.e. the energy Equation 24 is still applied during expansion until the opening of the exhaust valve.

The simulation model was applied to a direct-injection diesel engine: 79.5 mm bore, 106.2 mm stroke, 175 mm connecting rod, 16:1 compression ratio, 1500 rpm and (26.2 mg/cycle/cylinder) fuelling at full load. The inlet valve opens at 6 deg. BTDC and closes at 35 deg. ABDC. The exhaust valve opens at 40 deg. BBDC and closes 11 deg. ATDC. Details of the calculations procedure, computer programmes and subroutines are given in Reference 33. Samples of results and discussion are presented.

RESULTS AND DISCUSSION

Figure 1 shows the pressure-volume diagram for the gas exchange processes. In order to start computations of the intake process; the pressure and temperature of the residual gases (at TDC) are assumed to be 101 kPa and 600 K respectively. Actual values of pressure and temperature in the cylinder are determined after a complete cycle analysis. The valve overlap period from the inlet valve opening (6 degrees crank angles BTDC) to the exhaust valve closure (11 degrees crank angles ATDC) would determine the actual cylinder conditions for a new intake process calculations. This figure will be useful for a further study on the internal combustion engine pumping losses and turbocharging.

Figure 2 shows the changes in the cylinder pressure due to the effect of varying engine speed. Increasing engine speed will decrease the pressure during the intake process, i.e. decreasing the mass of the cylinder charge. Figure 3 shows the logarithmic plot of the pressure-volume diagram for the compression and expansion processes. This logarithmic plot is useful in determining the polytropic indices for the compression and expansion processes.

Figure 4 shows the changes in the cylinder pressure during the combustion process. The sudden rise in pressure is due to the premixed mode of combustion up to the peak cylinder gas pressure. Then the diffusion combustion completes the combustion process. Figure 5 shows the corresponding cylinder gas temperature behaviour during the combustion process.

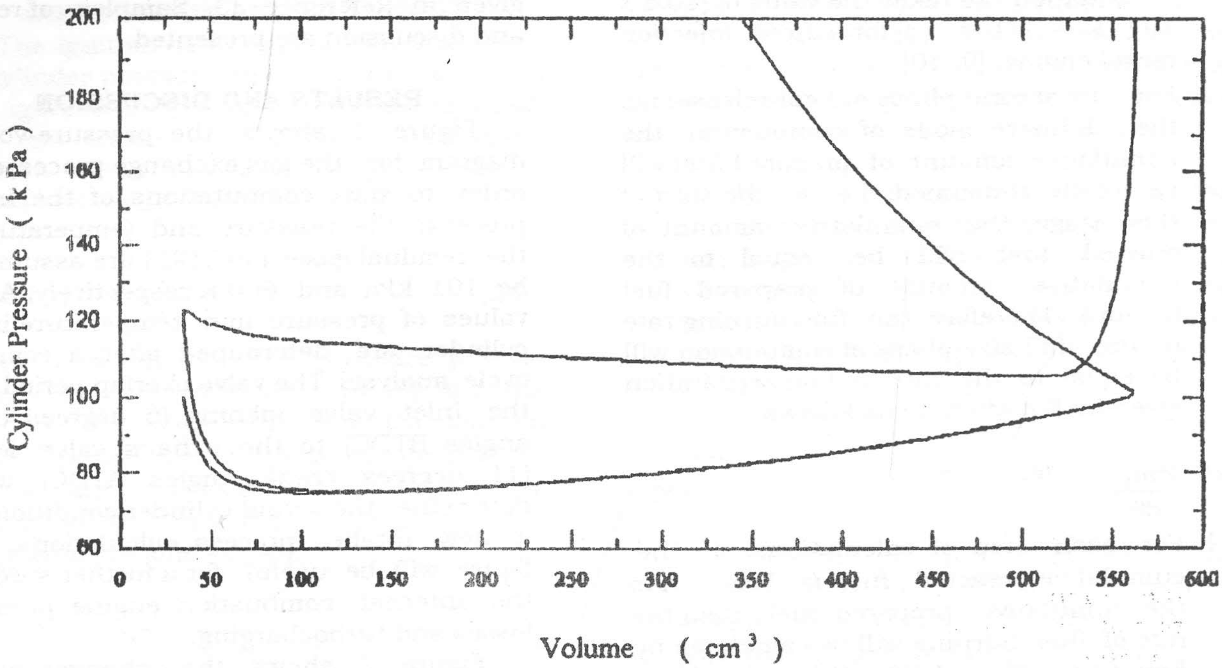

Figure 1 The pressure-volume diagram for the gas exchange processes; the intake process, the exhaust process and the valve overlap period. (the lower curve shows the start of cycle analysis computions at TDC with suitable assumptions).

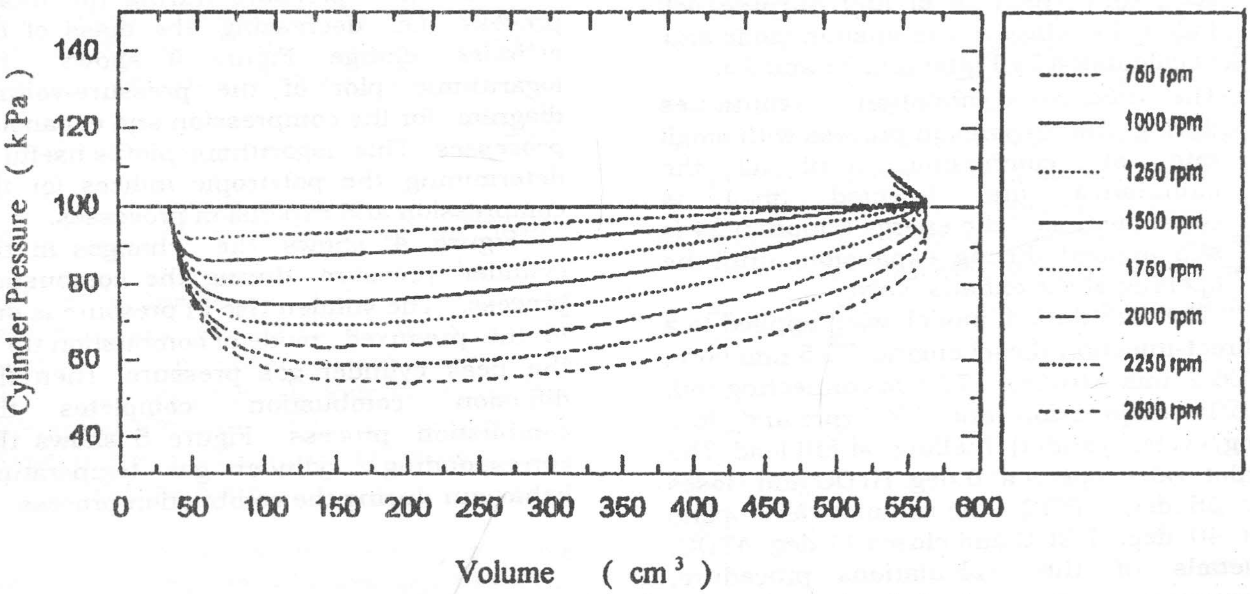

Figure 2 The effect of engine speed on the cylinder pressure during the intake process.

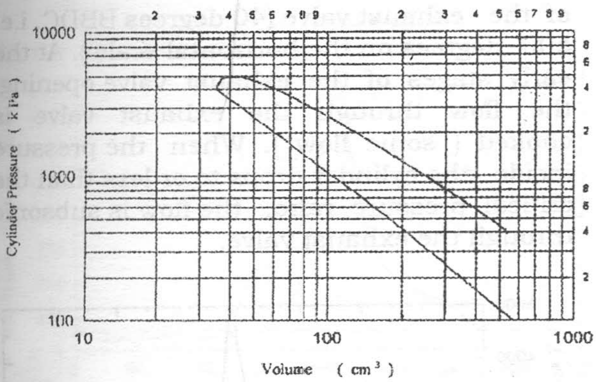

Figure 3 The logarithmic pressure-valume diagram for the compression and expansion processes.

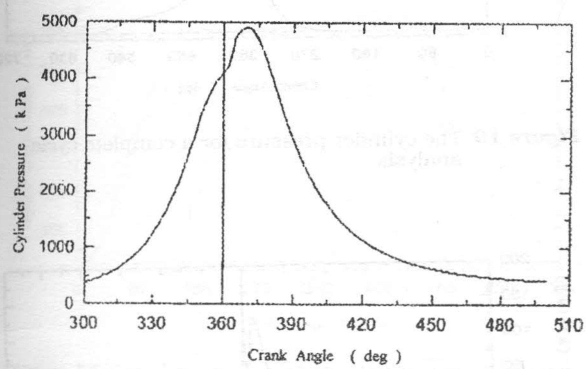

Figure 4 The effect of the combustion process on the= cylinder pressure.

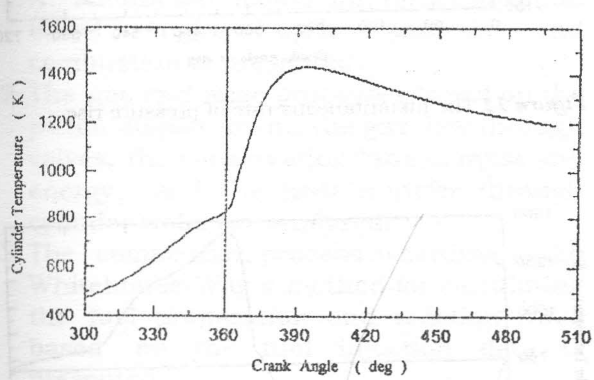

Figure 5 The effect of the combustion process on the cylinder temperature.

Figure 6 shows the fuel injection rate and its effect on the preparation and burning rates. It is noticed that the peak of the fuel preparation rate is reached at the end of the fuel injection, i.e. at the maximum amount of cumulative fuel injected. The rapid stage of the fuel burning, i.e. the premixed combustion, occupies few degrees crankangles, then the diffusion combustion completes the combustion process. Figure 7 shows the changes in the heat release pattern due to the change in engine loading. Figures 8 and 9 show the corresponding changes in cylinder pressure and temperature.

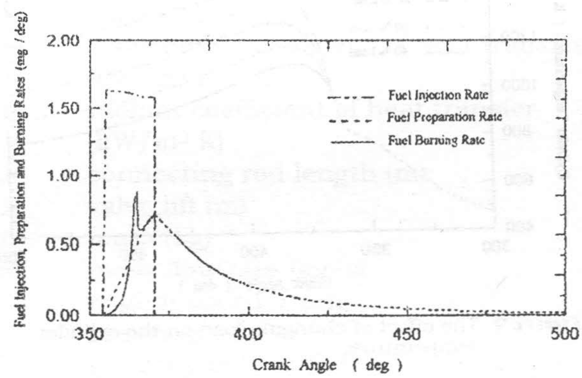

Figure 6 The fuel injection, preparation and burning rates for the full load case.

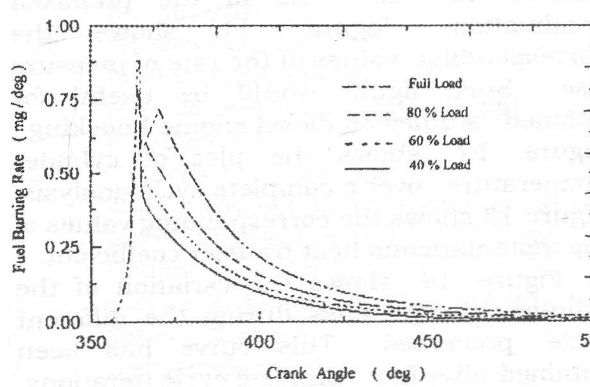

Figure 7 The fuel burning rates for various loads.

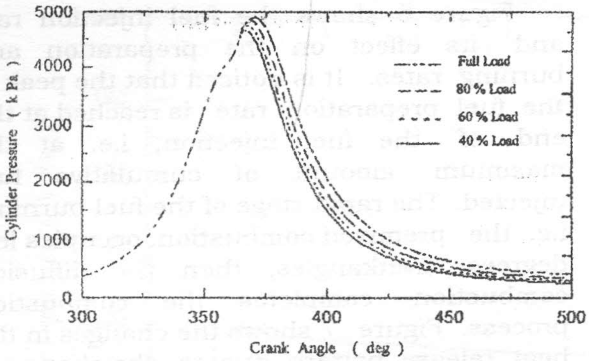

Figure 8 The effect of changing load on the cylinder pressure.

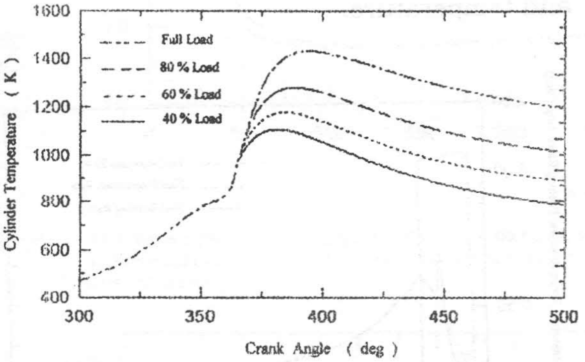

Figure 9 The effect of changing load on the cylinder temperature.

Figure 10 shows the plot of cylinder pressure over a complete cycle analysis. The rapid increase in cylinder pressure is noticed at the zone of the premixed combustion. Figure 11 shows the corresponding values of the rate of pressure rise. Such figure would be useful for detailed studies on diesel engine knocking. Figure 12 shows the plot of cylinder temperature over a complete cycle analysis. Figure 13 shows the corresponding values of the instantaneous heat transfer coefficient.

Figure 14 shows the variation of the cylinder charge mass during the different cycle processes. This curve has been obtained after four complete cycle iterations. Accurate determination of the mass of the residual gases can be obtained from this figure. The rate of fuel burning shows slight increase in cylinder charge mass during the combustion process. It is also noticed that the exhaust blowdown starts at the opening of the exhaust valve (40 degrees BBDC, i.e. 500 degrees on the horizontal scale). At the early stages of the exhaust valve opening, the flow through the exhaust valve is choked (sonic flow). When the pressure inside the cylinder drops to or less than the critical pressure ratio, the flow is subsonic through the exhaust valve.

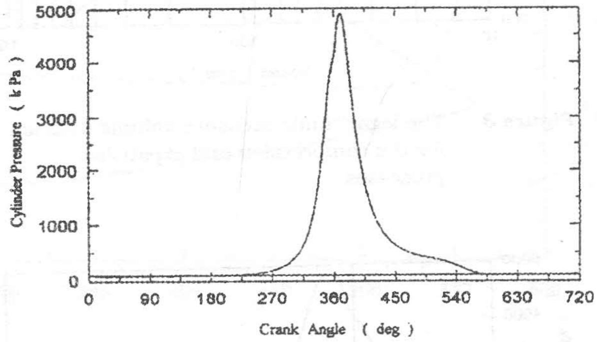

Figure 10 The cylinder pressure for a complete cycle analysis

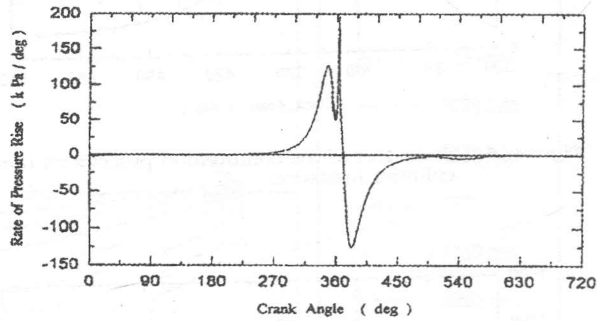

Figure 11 The instantaneous rate of pressure rise.

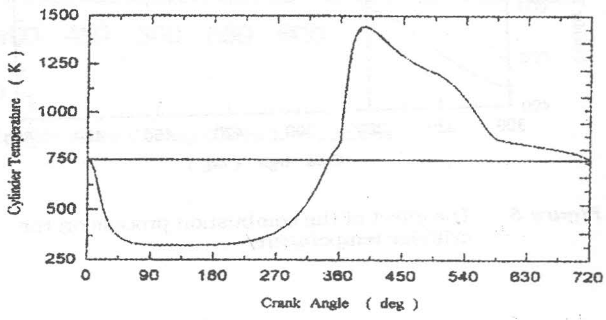

Figure 12 The cylinder temperature for a complete cycle analysis

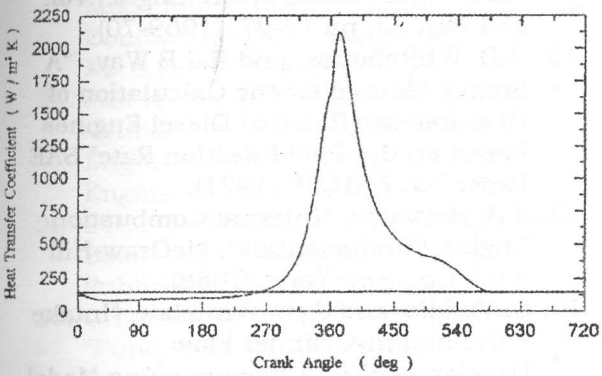

Figure 13 The instantaneous heat transfer coefficient

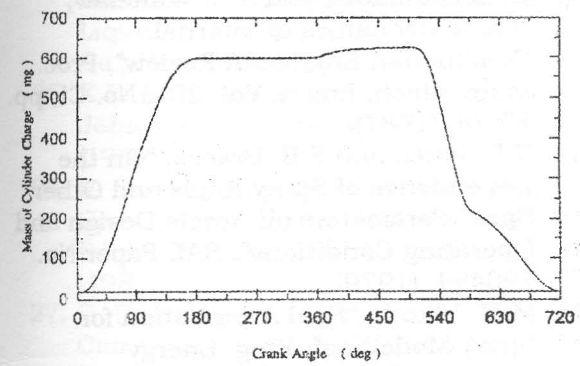

Figure 14 The change of cylinder charge mass during cycle processes

CONCLUSIONS

1. A simulation model for direct-injection diesel engine cycle analysis and combustion is presented.
2. The gas exchange processes based on the piston displacement, the gas flow through valves, the conservation laws of mass and energy, and the heat transfer through cylinder walls are analyzed.
3. The combustion process according to the Whitehouse-Way`s method for calculating the fuel preparation and reaction rates based on the fuel injection data is presented.

RECOMMENDATION

It is recommended, for future study, to add a submodel to the present work in order to study the diesel engine exhaust emissions especially nitrogen oxides and smoke.

NOMENCLATURE

A_c	curtain area (π D_v L_v) (m^2)
A_p	piston area (m^2)
A_R	reference area (m^2)
A/F	air to fuel mass ratio
BDC	bottom dead center
c_1, c_2	constants in Equations 26 and 28
c_p	specific heat at constant pressure (kJ/kg K)
c_v	specific heat at constant volume (kJ/kg K)
C_d	coefficient of discharge
CV	lower heating value (kJ/kg)
D_v	valve diameter (m)
h_c	convective coefficient of heat transfer (kW/m^2 K)
h_r	radiant coefficient of heat transfer (kW/m^2 K)
L_c	connecting rod length (m)
L_v	valve lift (m)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
N	engine speed (rpm)
p	pressure (kPa)
Q	quantity of heat (kJ)
r	crank radius (m)
R	gas constant (kJ / kg K)
$\overline{R}$	universal gas constant(kJ/kmole K)
t	time (s)
T	temperature (K)
TDC	top dead centre
U	total internal energy (kJ)
u	specific internal energy (kJ/kg)
V	volume (m^3)
V_c	clearance volume (m^3)
v_p	instantaneous piston speed (m /s)
x	piston displacement from TDC (m)
γ	specific heats ratio, (c_p/c_v)
λ	excess air factor, [$(A/F)_{act}$ / $(A/F)_{th}$]
ρ	density (kg/m^3)
θ	crankangle (deg)
τ	ignition delay time (ms)

Subscripts

a	air
b	burnt fuel
e	exit from the control volume
f	fuel

g	gas
i	inlet to the control volume
inj	injected fuel
noz	injector nozzle
o	stagnation property
p	prepared fuel
r	reacted fuel
u	unprepared fuel
w	cylinder walls

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نموذج محاكاة الإحتراق والدورة الحقيقية للمحرك الديزل

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ملخص البحث

البحث عبارة عن نموذج محاكاة أداء المحرك الديزل. وهذا النموذج الذى يشمل عملية الإحتراق والدورة الحقيقية للمحرك يعتبر ذو أهمية كبيرة بالنسبة لتصميم وتطوير محركات الإحتراق الداخلى. والبحث يحتوى على نماذج لكل من حركة المكبس وسرعته بالنسبة لزوايا عمود المرفق وعمليات تصرف الغازات خلال صمامات السحب والعادم. وتطبيق قوانين بقاء المادة والطاقة وإستخدام علاقات إنتقال الحرارة خلال جدران المحرك وحساب معدل إحتراق الوقود بإستخدام طريقة (هوايت هاوس - واى). ولتحقيق درجة عالية من الدقة قسمت حركة عمود المرفق إلى فترات تراوحت بين (٢-٥) درجات خلال عمليات السحب والعادم ولكنها خفضت إلى أقل من درجة واحدة خلال مرحلة الإحتراق السريع. ويشتمل البحث على نتائج للعمليات المختلفة لدورة المحرك ومنها عمليات تبادل الغازات (السحب والعادم) ومعدلات تحضير وحرق الوقود ومنحنيات كل من الضغط ودرجة الحرارة ومعامل إنتقال الحرارة اللحظى مع زوايا عمود المرفق وكذلك معدل إنبعاث الحرارة أثناء عملية إحتراق.