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A Study Of Combustion Temperature Distribution In The Cylinder Of Compression Ignition Engine

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ABSTRACT:*This paper investigates the combustion temperature distribution and consequent heat transfer to the wall of the combustion chamber of a compression ignition engine. The gas temperature and pressure were obtained using a model, and the model was analyzed using Matlab 8.5a programming language, by varying engine speed at various crank angles. The Arrhenius equation for chemical kinetic model was utilized and combustion analyzed for constant air/fuel and compression ratio. In addition, the Annand and Ma heat transfer model was used to determine the heat loss to the cylinder wall during combustion. At engine of 2000rpm and crank angle of -15 to 15, the temperature and heat obtained were 2300K and 2300kJ receptively. At engine speed of 3000rpm and crank angle of -15 to 15 degrees, the temperature and heat obtained were 3150K and 3150kJ respectively. At engine speed of 4000rpm and crank angle of -15 to 15, the temperature and heat obtained were 3970K and 3970kJ respectively. The results obtained from the research revealed the dependency of combustion temperature distribution on engine speed. The combustion temperature distribution and heat transfer to the wall of the combustion chamber were observed increasing with engine speed, reaching a maximum immediately after the TDC and then decreased sharply due to expansion stroke as a result of the opening and closing of the inlet and exhaust valves. This research will help in assessing the performance level of a reciprocating internal combustion engine and will help designers to design for thermal stress reduction in engine, and provide sufficient path of heat flow to the various parts of the engine to ensure that temperature at such parts do not exceed safe values if implemented.*

Keywords: *Combustion-temperature; in-cylinder-heat-transfer; compression-ignition-engine; engine-speed; Crank-angle*

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I. INTRODUCTION

Compression-ignition engines popularly known as direct injection (DI) diesel engines are said to be the most used and efficient prime movers commonly available today [1]. The importance of compression-ignition engines makes it necessary to be studied and investigated for further improvements to satisfy human needs. Diesel engine moves a large portion of the words equipment and it plays a vital role in transportation of goods and passengers on the road and high sea. It is expected that diesel engine will be in use for another hundred years as increasingly economical sources are found with the increase in oil prices offering incentive to the explorer [1].

The investigation of heat transfer in the internal combustion engine is an important issue because critical engine parameters such as in-cylinder temperature and pressure is affected by heat, therefore a good knowledge of combustion and flow in the internal combustion engine is required for improving the engines performance. It is eminent that multidimensional mathematical models are powerful tools for investigating flow, temperature distribution, influence of engine speed on the temperature distribution and understanding complex systems which includes combustion process of compression-ignition engines [2].

[3]Modified a single-cylinder direct injection diesel engine to operate on compression ignition of homogeneous mixture of diesel fuel and air. The research suggests that controlled homogeneous charge compression ignition (HCCL) is achievable and EGR rate, compression ratio, and air fuel rate are practical controlling factors in achieving satisfactory operation. [4]Developed a calculation method based on the control volume approach for solving two-dimensional elliptic problem involving fluid flow, heat and mass transfer.

Although the method is described for steady two-dimensional situations, its extension to unsteady flow and three- dimensional problems is very straight forward.

[5] Carried out a combined experimental and theoretical investigation of the phenomenon of short term response (cyclic) temperature transients in the combustion chamber walls of a reciprocating internal combustion engine. The experimental work was carried out on a direct injection (DI), air cooled, four-stroke, diesel engine. The influence that engine operating conditions and wall material properties have on the values of cyclic temperature swings was demonstrated.

[6] Explained that knowledge of piston and cylinder wall temperature is necessary to estimate the thermal stresses at different points; this gives an idea to the designer to take care of the weaker cross sectional area. The proposed methodologies have been successfully applied to a water-cooled four-stroke direct-injection diesel engine and it allows the estimation of the piston and cylinder wall temperature.

The aim of this paper is to simulate a model for the prediction of combustion temperature distribution in the cylinder of compression ignition engine. The objectives are to numerically investigate the influence of engine speed on combustion and heat transfer for constant air/fuel ratio. Also, to assess the performance level of reciprocating internal combustion engine.

II. MATERIAL AND METHODS

2.1Methodology

The basic formulations employed in this study is introduced. The governing equations for reacting flow gas mixtures will be expressed in a differential form. It covers a detailed review taking cognizance of the cases of laminar reacting flows which is the flow of fluid when each particles of the fluid follows a smooth path and turbulence flow which is an irregular flow of fluid which is in contrast to laminar flow. Useful information which includes the in-cylinder bore, initial wall temperature, engine stroke, engine speed, piston crown, compression ratio will be obtained from the manufacturer's engine specification and calculated for proper analysis. A two- dimensional model shall be used for predicting or simulating the combustion chamber wall temperature of a compression ignition (C I) engine.

2.2 Combustion Model Equation

The Arrhenius equation for chemical kinetics as established by [7] is given by the following equation:

$$\rho c_p \frac{\partial T}{\partial t} + = \mathrm{Div} + (\mathrm{kgradT}) \;\; \rho c_p v_p \mathrm{gradT} - \rho c_p v_p \mathrm{gradT} - \mathrm{gradT} + \; QR_f \qquad (1)$$

In an axi-symmetric cylinder, equation (1) simplifies to:

$$\rho c_p \frac{\partial T}{\partial t} + = \mathrm{Div} + (\mathrm{kgradT}) \;\; \rho c_p v_p \mathrm{gradT} - \rho c_p v_p \mathrm{gradT} - \mathrm{gradT} + \; QR_f \qquad (2)$$

where,
p is the cylinder gas density
c_p is the specific heat capacity of gas mixture at constant pressure, k is the thermal conductivity
$\rho c_p v_p$ are the radial and axial velocities of the fluid
Q is the heat of reaction which is a function of the type of fuel used
R_f is the volumetric rate of heat generation of species in the cylinder already defined as:

$$R_f = AP^2 \omega_f{}^a \omega_w{}^b \exp\left[\frac{E_a}{R_u T}\right] \qquad (3)$$

A is the pre-exponential factor
And w_{fo} are unburned fuel and oxidizer mass fractions
a,b are constants
E_aIs the activation energy which is also a fuel parameter.
R_u is the universal gas constant and
T is the temperature
In order to be able to determine the temperature distribution in a medium, the medium is taken to be stationary and hence the equation (2) describing the combustion in the medium simplifies to:

$$\rho c_p \frac{\partial T}{\partial t} + = Div + (kgradT) + +QR_f \qquad (4)$$

Substituting for R_f into equation (4) yields

$$\rho c_p \frac{\partial T}{\partial t} + = Div + (kgradT) + + QA_\rho{}^2 \omega_f \omega_{ox}{}^b \exp\left[-\frac{E_0}{R_u T}\right] \tag{5}$$

The divorce of the gradient of temperature with a constant conductivity in a cylindrical coordinate. first term of the right hand side of equation (5) could be further expanded as follows:

$$kDiv(gradT) = k\nabla^2 T = k\frac{1}{r}\left[\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \right] + \frac{1}{r^2}\frac{\partial}{\partial \xi}\left(\frac{\partial T}{\partial \psi}\right) + \frac{\partial}{\partial z}\left(\frac{\partial T}{\partial z}\right) \tag{6}$$

From the assumption that the variation of variables along the azimuthal angle is not of interest in an axisymmetric cylinder, equation (6) reduces to:

$$kD^2 T = k\left[\frac{1}{r}\left(\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)\right) + \frac{\partial}{\partial z}\left(\frac{\partial T}{\partial z}\right)\right] \tag{7}$$

where,

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = \frac{1}{r} * r * \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} * \frac{\partial T}{\partial r} * \frac{\partial r}{\partial r} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} \tag{8}$$

On substituting into 5 we shall have:

$$\rho c_p \frac{\partial T}{\partial t} = k\left[\frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2} + \frac{\partial^2 T}{\partial r^2}\right] + QA_\rho{}^2 \omega_f \omega_{ox}{}^b \exp\left[-\frac{E_0}{R_u T}\right] \tag{9}$$

Since the engine parameters are often investigated with reference to time or crank angle intervals as a better alternative, as the piston moves from the top dead center to the bottom dead center, the above equation is converted from a time domain to a crank angle domain using the following relation.

$$\theta = \omega t \tag{10}$$

where

θ is the crank angle in radians

ω is the angular velocity of the crank shaft in radians per second

t is the time in seconds

Differentiating equation 10 with respect to time yields

$$\frac{\partial \theta}{\partial t} = \omega \tag{11}$$

Given that the angular velocity of the crank shaft ω is a function of engine speed which is

$$\omega = 2\pi N \tag{12}$$

Where N is the engine speed in revolutions per minute, then equation 11 becomes:

$$\frac{\partial \theta}{\partial t} = 2\pi N = 6.284N \tag{13}$$

Rearranging, equation 13 for ∂t and substituting into equation 9, we will obtain:

$$\frac{\partial \theta}{\partial t} = \frac{\alpha}{6.284N}\left[\frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2} + \frac{\partial^2 T}{\partial r^2}\right] + \frac{QA_\rho{}^2 \omega_f \omega_{ox}{}^b \exp\left[-\frac{E_0}{R_u T}\right]}{6.284N\rho c_p} \tag{14}$$

Sources: [7]

where;

$\alpha = \dfrac{k}{\rho c_p}$ is the thermal diffusivity of the reacting species

Equation 14 is therefore the governing combustion differential equation in an axi-symmetric cylinder and crank angle domain. This becomes the equation to be utilized in determining the temperature distribution in the cylinder of an internal combustion engine during the combustion period usually occurring between -15 degrees before top dead centre (TDC) and +15 degrees after top dead centre.

2.3 Non-Dimeinsional Equation of Combustion

In carrying out the simulation of combustion in the cylinder of an internal combustion engine, it is common to express the governing equations in dimensionless form for easy definition of parameters numerically yet retaining the accuracy of the computation considerably.
Indimensionalizing equation 14, the following dimensionless parameters are defined:

$r* = \frac{r}{r_0}$ as the dimensionless radial distance (15)

$z* = \frac{T}{L}$ as the dimensionless axial distance (16)

$\theta* = \frac{\theta}{0_0}$ as the dimensionless crank angle (17)

$T* = \frac{T}{T_{wl}}$ as dimensionless temperature (18)

where,

$r_0 = \frac{bore}{2}$

L = engine stroke

θ_0 is the maximum crank angle as the piston moves from TDC to BDC which is 180^0

T_{od} is the maximum flame temperature (or adiabatic flame temperature) of the fuel used in the simulation .

On substituting equations (15) to (18) appropriately into (14) shall have:

$$\frac{\partial(T*T_{od})}{\partial(\theta*\theta_o)} = \frac{\alpha}{6.284N}\left[\frac{1}{r}\frac{\partial T}{r_0} + \frac{\partial(T*T_{od})}{\partial(r*r_o)} + \frac{\partial^2(T*T_{od})}{\partial(r*r_o)} + \frac{\partial^2(T*T_{od})}{\partial(z^{*2}L^2)}\right] +$$

$$\frac{QA_\rho{}^2\omega_f\omega_{ox}{}^b\exp\left[-\frac{E_0}{T_{od}R_uT}\right]}{6.284N\rho c_p} \tag{19}$$

Factoring out the sealing parameters simplifies equation 19 to the following expression:

$$\frac{\partial(T*T_{od})}{\partial(\theta*\theta_o)} = \frac{\alpha}{6.284N}\frac{\theta_0}{r_0{}^2}\left[\frac{1}{r*}\frac{\partial T*}{\partial r*} + \frac{\partial^2 T*}{\partial r*^2}\right] + \frac{\alpha\vartheta_0}{6.284NL^2}\frac{\partial^2 T*}{\partial z*^2} + \frac{QA_\rho{}^2\omega_f\omega_{ox}{}^b\exp\left[-\frac{E_0}{T_{od}R_uT*}\right]}{6.284N\rho c_pT_{ad}} \tag{20}$$

Equation 20 is therefore the second-order non-linear combustion equation in a dimensionless form in an axi-symmetric cylinder and should be applied in the multi-dimensional simulation of combustion in the cylinder of an internal combustion engine of any configuration.

2.4 In-Cylinder Heat Transfer Model

The heat generated during combustion depends on the dimension of the cylinder bore, the type of fuel, air-fuel ratio, and lower heating value of the fuel that is used which is typically equal to the enthalpy of reaction of heat of reaction in the opposite sign. Usually the heat generated by combustion is called the heat addition. The heat transfer by convection is:

$q = \boldsymbol{\beta\sigma}(T_g^4 - T_w^4)$ (21)

Where T_g and T_w are the gas temperature and wall temperature respectively.
The heat transfer to the containing chamber walls is modeled using [8]correlation, given as:

$$q = \frac{Q_w}{A_w} = q^{cv} + q^R = c\frac{k}{B} R_e^b \left(T_g - T_w\right) + \left(T_g^4 - T_w^4\right) \tag{22}$$

[9]suggested that for a compression ignition engine:
c is a constant in the range 0.25<c<0.8
b = 0.7
β = 0.576
σ is the Stefan-Boltzmann constant
R_0 is the flow Reynolds number expressed as:

$$R_0 = \frac{\rho S_\beta B}{\mu} \tag{23}$$

where;
ρ isdensity (kg/m^3)
B isthe cylinder bore in (m)
μ is the gas dynamic viscosity in (kg/ms), and
S_β isthe mean piston speed as:

$$S_p = 2LN \tag{24}$$

L is the engine stroke (m), and
N is the engine speed in (rpm)

Motored cylinder pressure, $P_m = P_{IVC}\left(\frac{V_{IVC}}{V}\right)^r$ (25)

Gas-side convective heat transfer coefficient

$$\frac{Q_w}{A} = h_{eg}\left(T_g - T_w\right) + \beta\sigma\left(T_g^4 - T_w^4\right) \tag{26}$$

2.5 Chemical Kinetics Model

The stoichiometric combustion reaction is expressed as follows:

$$C_{12}H_{26} + a(0_2 + 3.76N_2) \rightarrow bCO_2 + CH_2 0 + 3.76aN_2 \tag{27}$$

$$\left.\begin{array}{l}\text{Thus for:}\\ \text{Carbon: b-12}\\ \text{Oxygen: 2a = 2b+c=24+c}\\ \text{Hydrogen: } 26=2c \Rightarrow a=18.5\\ \text{Substituting equation 28 into 27 yields:}\end{array}\right\} \tag{28}$$

$$C_{12}H_{26} + 18.5(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 18.5\,(3.76)N_2 \tag{29}$$

3.8 Engine Parameters for the Analysis

The fuel used is n-Dodecane ($C_{12}H_{26}$) which approximates the practical light diesel fuel. The initial fuel temperature was taken as 300k and the model engine specifications are listed in Table-3.1 below:

Table 1: Specifications of Model Engine [10]

S/N	PARAMETER	SYMBOL	UNIT	VALUE
1.	Cylinder bore	B	M	0.120
2.	Engine stroke	L	M	0.153
3.	Piston crown			Flat
4.	Engine speed	N	RPM	2000,3000,4000
5.	Compression ratio			15.0:1
6.	Fuel			n-Dodecane
7.	Initial fuel temperature	Tr	K	300
8.	Duration of combustion			-15^0BTDC to 15^oATDC
9.	Crank radius	A	M	0.062
10.	Connecting rod length	L	M	0.21
11.	Crank throw	CT	M	0.0575
12.	Engine swept volume	V_S	M^3	8.153 X $^{10-4}$
13.	Clearance volume	V_C	M^3	4.94 X 10^{-5}
14.	Connecting rod length/crank radius	R_S		3.4
15.	Expansion index	R		1.33

The crank radius, connecting rod length, crank throw, engine swept volume, clearance volume, and connecting rod length/crank radius ratio are determined from a basic geometric analysis of the engine components [7], is summarized above.

III. RESULTS AND DISCUSSION

The results were obtained by modeling the respective equations using matlab 8.5a programming language. Using N = 2000 [rpm], 3000 [rpm], 4000 [rpm], $p_1 = 0.6x\ 10^5\ [Nm^2], T_1$=300[K], the following results were obtained at various crank angles and engine speeds.

3.1 Variation of in-cylinder gas temperature against crank angle for different engine speeds

Figure 1 shows the variation of in-cylinder gas temperature against crank angle for different engine speeds. According to the equation of state for the ideal gas, temperature is a function of the pressure. Gas temperature variation inside the cylinder is identical pattern with the pressure profile. For the compression stroke, the in-cylinder gas temperature increases continuously from about 167k to a maximum of about 2300k, which correlate with an engine speed of 2000rpm. However, for an engine speed of 3000rpm, the temperature increased from about 950k to about 3150k. Same is applicable for 4000rpm and subsequent rpm values with coincident behaviour for the in-cylinder pressure. Behind the end of compression stroke (TDC) the in-cylinder gas temperature increases rapidly until the peak temperature location as a result of combustion occurrence after TDC. However, the in-cylinder gas temperature starts to decrease during expansion stroke from about 2300k to the initial temperature of about 167k for an engine speed of 2000rpm. Also, for an engine speed of 3000rpm the temperature dropped from about 3150k to the initial temperature of about 950k. same is applicable for 4000rpm and subsequent rpm values, but have smaller gradient compared with in-cylinder pressure for the same period. The temperature drop is continuous for other cycle strokes (power and exhaust) with the same gradient for exhaust stroke whilst rapidly gradient at the beginning of intake stroke due to opening of intake valve and thus a fresh cool charge enters into the cylinder. All of these happens within the crank angle of -15 to 15 degrees. It can be seen that the maximum temperature is obtained at higher engine speed; however, the difference is clearer after TDC during the combustion period. Peak temperature values appears after TDC due to start of combustion at TDC and continues the combustion process till the use of all fuel inside the cylinder.

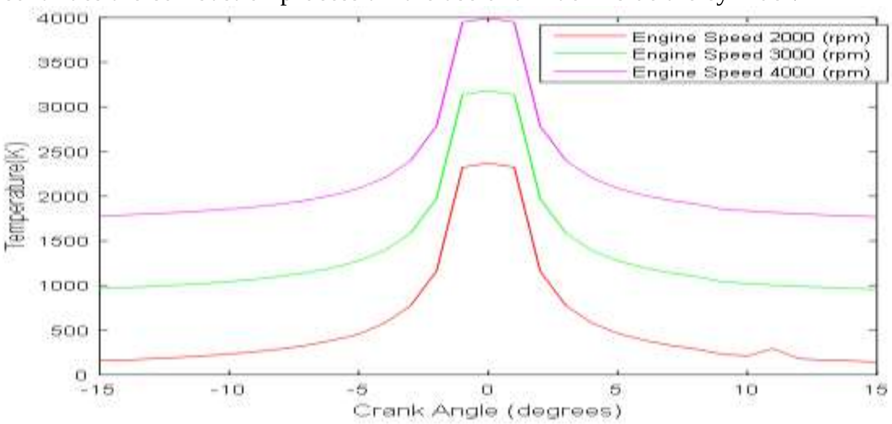

Fig 1 Variation of in-cylinder gas temperature against crank angle

3.2 Variation of motor pressure against crank angle for different engine speeds.

Figure 2 shows the variation of motor pressure against crank angle with different engine speeds. The pressure increases from about 1.67bar to a maximum of about 26bar and progressively decreases from about 26bar to the initial pressure of about 1.6bar for an engine speed of 2000rpm. However, for an engine 3000rpm the pressure increased from about 11bar to a maximum of about 35bar and dropped to initial pressure after TDC. Same is applicable for 4000rpm and subsequent rpm values. All of these happens within crank angle of -15 to 15 degree. It can be seen that pressure increases with increase of engine speed and the pressure drops dramatically immediately after TDC.

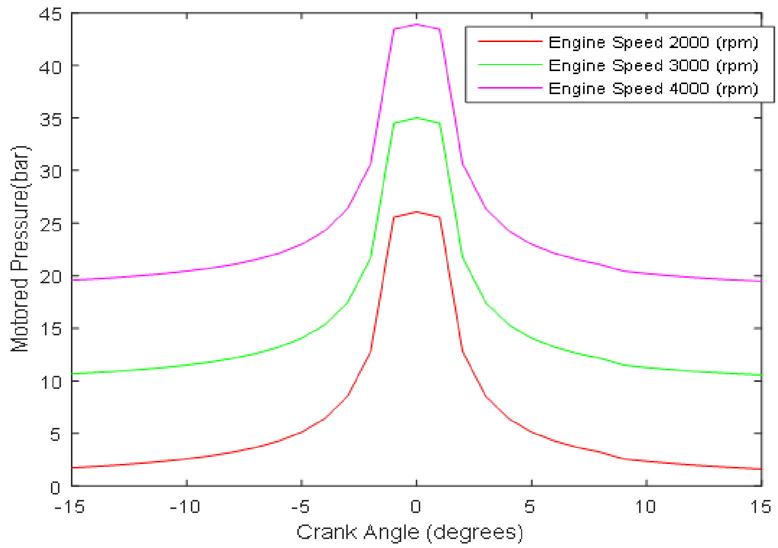

Fig 2 Variation of in-cylinder gas motor pressure against crank angle

3.3 Variation of heat transfer coefficient against crank angle for different engine speed

Figure 3 show that the heat transfer coefficient is increasing from about 167kJ to a maximum about 2300kJ and progressively decreases to initial heat transfer coefficient for an engine speed of 2000rpm. However, for an engine 3000rpm the heat transfer coefficient increased from about 950kJ to a maximum of about 3150kJ and dropped to the initial heat transfer coefficient of about 950kJ. Same is applicable for 4000rpm values and subsequent rpm values as the engine speed increases due to increasing the driving force for the heat transfer process (forced convection).

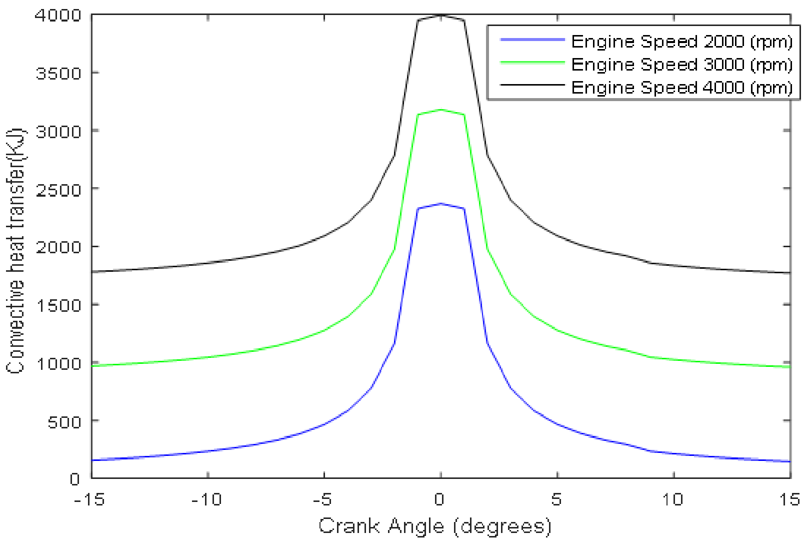

Figure 3 Variation of in-cylinder gas heat transfer coefficient against crank angle

IV. CONCLUSION

A multidimensional simulation of combustion process and heat flow in the compression ignition engine fueled by hydrocarbon fuel have been done using Anand and Ma heat transfer model and MATLAB code to simulate the process numerically. The program written from the simulation process can aid engine designers in the design of internal combustion engines for alternative fuels. Also, it can used to study other problems of interest.

It is concluded that the temperature distribution is dependent on the engine speed, and that the heat transfer in the cylinder wall has shown a decreasing tendency as the piston moves from TDC to BDC along the cylinder axis. Hence, accurate knowledge of the temperature distribution in the engine cylinder, mostly during the combustion period will in no mean manner aid the prediction of effective cooling of engine components and invariably ameliorate the emissions of some poisonous combustion products such as No_x.

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NOMENCLATURE

μ	gas dynamic viscosity	$\left\{\frac{kg}{ms}\right\}$
K	thermal conductivity	$\left\{\frac{W}{mK}\right\}$
A	thermal diffusivity	$\left\{\frac{K}{\rho c\rho}\right\}$
Θ	crank angle	{degrees}
ω	angular velocity of the crank shaft	{rps}
ρ	cylinder gas density	$\left\{\frac{kg}{m^3}\right\}$
ϱ	heart of reaction	$\left\{\frac{kJ}{kmol}\right\}$
R_r	volumetric rate of heat generation of species	
T	time	{s}
U_f	radial velocity component	{m/s}
U_s	wall velocity component	{m/s}
A	pre-exponential factor	$\{cm^3 / kmol\}$
N	engine speed	{rpm}
E_a	activation energy	{K}/{kmol}
CR and CR^1	combustion model constants	
W_f	fuel mass fraction	
W_{ox}	oxidizer mass fraction	
R_u	universal gas constant	{kj/kmolK}

V_s	engine swept volume	{m^3}
‘α	crank radius	{m}
T_{od}	maximum flame temperature	{K}
L	engine stroke	{m}
h_c	convective heat transfer coefficient	{W/m^2 k}
T_g	gas temperature	{K}
T_w	wall temperature	{K}
R_e	Reynolds number	
B	cylinder bone	{M}
S_p	mean piston speed	
r_c	compression ratio	
V_c	clearance volume	
R_s	connecting rod length/crank radius ratio	

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