

# Development of Four-Zone Segmented Transitional Model for Reciprocating Internal Combustion Engine Analysis Using Gasoline

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**Abstract:** A four zone model based on the first law of thermodynamics has been developed for analysis of combustion in an internal combustion engine. The four zones included an unburned zone and two regions of burned zone, (namely burned gas 1 and burned gas 2) and unburned burned zone described as a transitory zone which is a mixture of burned and unburned gases. Arbitrary constant for each of burn (CC2) and unburned (CC1) zone leakages in unburned burned zone was evaluated at optimally predetermined values of 0.005 and 0.00025 respectively, while mass fraction burned from burned gas 1,  $x_1$  and burned gas 2,  $x_2$  were also evaluated at predetermined optimal values of 0.6 and 0.4 respectively. The model was used to analyse an SI engine operating with a gasoline fuel. The engine operating conditions were set at engine speed of 2000 rpm, -35bTDC ignition time and burn duration at 60°. The temperature distribution from the arbitrary constants (CC2, CC1,  $x_1$  and  $x_2$ ) for the newly developed four zone model was compared to the two zone model and literature experimental temperature value. The obtained indicated mean effective pressure (IMEP), thermal efficiency ( $\eta$ ), cylinder pressure and emission characteristics from the developed model and those of two zone analysis were both compared with literature values.

**Keywords:** Zones, Burn Duration, Ignition Delay, Spark Ignition Engine, Combustion, Emission

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## 1. Introduction

The Spark-Ignition (Otto engines, gasoline and petrol engine) and the compression ignition (diesel engine) are the common examples of ICEs [1]. In internal combustion engine ICEs energy is released by burning or oxidizing fuel inside the engine. The fuel air mixture before combustion and the burned products after combustion are the actual working fluids. Basic mechanism of ICE has not changed till date since its invention. Major difference between modern day engine and ones built ages ago are improved reliability, thermal efficiency and emission level. Through the years, researches on ICE were directed and are still aimed towards improving thermal efficiency, emission rate and noise reduction as well as vibration. Thermal efficiency has increased from 10 to 50% today while emission is 5% of what it was 40years ago [2]. The internal combustion engine

is being used widely, variety of emission it produces and effects have being discussed by many researchers, [3-7].

With models developed for each of the processes involved in engine cycle operation, basically namely; compression, combustion, expansion, simulation of a complete engine process can be built up which can be analysed to provide information on engine performance. Combustion of charge is one of the processes that controls engine power, efficiency and emissions [1]. In initial study of the combustion process, the combustion chamber is divided into two imaginary zones, burned ( $b$ ) and unburned ( $u$ ), [8, 9], but sometimes the estimates obtained from this analysis are not accurate. This has necessitated continuous efforts at developing some other multi-zone models in order to improve the emission and efficiency predictive capabilities. Some efforts at studying the combustion process tried to divide the combustion chamber to a number of zones, [10]. Further effort is being

made in the use of simulation model to achieve new engine ideas and technological advancement by increasing the number of zones being analysed.

In the present study, a four zone model was developed for performance analysis, see Figure 1 for model geometry. The model which had unburned zone consisting of the fresh charge, volume of which is represented as  $V_u$ , mixture of burned and unburned gases consisting of infiltration from both unburned and burned zones,  $V_{un}$  and partitioned burned zones into burned zone1,  $V_{b1}$  and burned zone2,  $V_{b2}$ , by virtue of uneven mixture proportion in the chamber leading to use of varying mass fraction rate of each burned zone and also varying the heat transfer coefficient to get the pressure, temperature, indicated mean effective pressure, thermal

efficiency and emission fraction readings.

## 2. Method

### 2.1. Mathematical Model

The 4-zone thermodynamic model being developed for this study consisting of unburned, two burned zones and transitory zones is shown in Figure 1. The burned zones divided into burned zone1 and burned zone2 by virtue of uneven mixture proportion in the chamber, while the transitory zone is formed by infiltration from the unburned and burned zones.

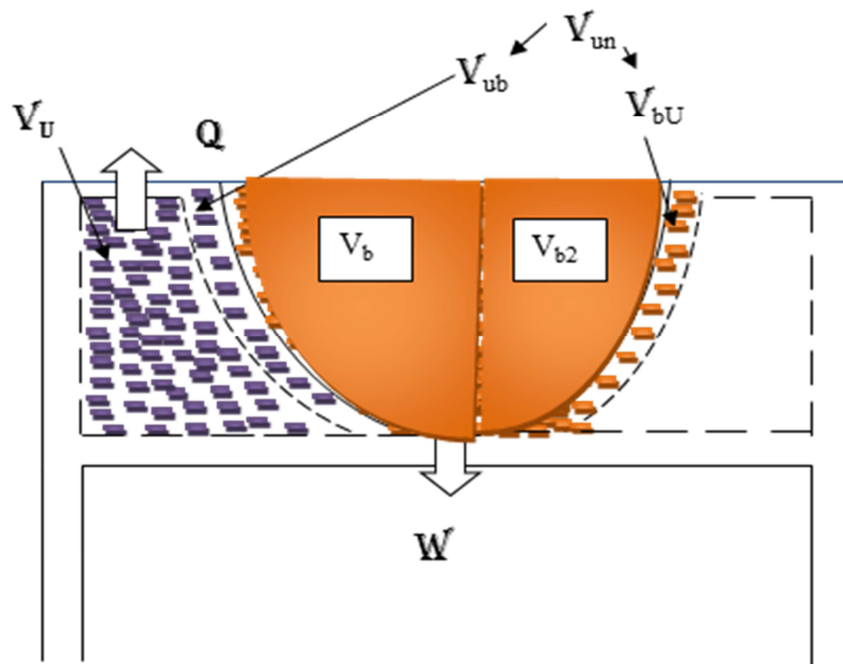


Figure 1. Geometry Representing the Four Zones (unburned zone, Unburned burned zone and burned zone1 and Burned2 Zone) In Combustion Chamber.

$V_{un}$  is the addition of volume of unburned gases in unburned burned zone,  $V_{ub}$  and volume of burned gases in unburned burned zone,  $V_{bu}$  that is,  $v_{un} = v_{ub} + v_{bu}$

The model equations are developed by applying the first law of thermodynamics to each zone as detailed below.

The rate of change of mass within any open system is the net flux of mass across the system boundaries, i.e.

$$m = \sum_k \dot{m}_k \tag{1}$$

The mass flux in the combustion chamber at any instant can be expressed as

$$m = m_u + m_b + m_{b2} \tag{2}$$

where,  $m_u$  is mass of unburned mixture,  $m_b$  is mass of burned gas,  $m_{b2}$  is mass of infiltration from burned zone and  $m_{un}$  is mass of mixture of burned and unburned gases.

Based on the application of the first law of thermodynamics to an open system, the energy equation is written as,

$$\dot{E} = \dot{Q} - \dot{W} + \sum \dot{m}h \tag{3}$$

where,  $E$ ,  $Q$  and  $W$  are internal energy of the cylinder mixture, heat exchange of content with cylinder wall and work done on piston by cylinder charge respectively. Equation (3) is thus applied to each of compression, expansion and combustion phases of the internal combustion engines as follows:

#### 2.1.1. Compression and Expansion Processes

For the combustion and expansion phases, eqn. (3) is transformed to

$$\frac{d(mu)}{d\theta} - \frac{dQ}{d\theta} - p \frac{dv}{d\theta} + h \frac{dm}{d\theta} \tag{4}$$

Now

$$\frac{d(mu)}{d\theta} = \frac{mdu}{d\theta} + \frac{udm}{d\theta} = m \left( \frac{du}{dT} \cdot \frac{dT}{d\theta} \right) + \frac{udm}{d\theta} \tag{5}$$

Thus by suitable manipulation

$$\frac{dT}{d\theta} = \frac{I}{m_c v} \left[ \frac{dQ}{d\theta} - p \frac{dV}{d\theta} + \frac{dm}{d\theta} RT \right] \quad (6)$$

By employing the ideal gas equation,  $PV = mRT$ , then

$$\frac{v dp}{d\theta} + p \frac{dv}{d\theta} = RT \frac{dm}{d\theta} + mT \frac{dT}{d\theta} + mR \frac{dT}{d\theta} \quad (7)$$

The cylinder composition is assumed frozen during compression and expansion processes i.e.  $\frac{dR}{d\theta} \cong 0$

$$\frac{dP}{d\theta} = \frac{1}{v} \left[ \frac{dm_l}{d\theta} RT + mR \frac{dT}{d\theta} - p \frac{dv}{d\theta} \right] \quad (8)$$

### 2.1.2. Combustion Process

During the combustion process, there will be three zones namely unburned, burned and infiltration from burned zone in the cylinder. The energy equation (Eqn. 3) is applied to each separately as detailed below

(i) Unburned zone

The energy equation for this zone is given as

$$m_u c_{v,u} \frac{dT_u}{d\theta} = -\frac{dQ_u}{d\theta} - p \frac{dV_u}{d\theta} - R_u T_u \frac{dm_x}{d\theta} - R_u T_u \frac{dm_{lu}}{d\theta} - R_u T_u \frac{dm_{ub}}{d\theta} \quad (12)$$

From the ideal gas equation, the following expression can be written

$$p \frac{dV_u}{d\theta} + \frac{V_u dP}{d\theta} = R_u T_u \frac{dm_u}{d\theta} + m_u R_u \frac{dT_u}{d\theta} \quad (13)$$

By substituting eqn. (13) in eqn. (12) yields

$$\frac{dT_u}{d\theta} = \frac{I}{m_u c_{p,u}} \left( -\frac{d\theta_u}{d\theta} + V_u \frac{dP}{d\theta} \right) \quad (14)$$

(ii) Burned zone 1

The energy equation for this zone is given as

$$m_b c_{v,b} \frac{dT_b}{d\theta} = -\frac{dQ_b}{d\theta} - p \frac{dV_b}{d\theta} + \frac{h_u dm_{x1}}{d\theta} - u_b \frac{dm_{x1}}{d\theta} - R_b T_b \frac{dm_{lb}}{d\theta} - R_b T_b \frac{dm_{bu}}{d\theta} \quad (16)$$

The ideal gas equation for this zone is expressed as

$$p \frac{dV_b}{d\theta} + V_b \frac{dP}{d\theta} = R_b T_b \frac{dm_b}{d\theta} + m_b R_b \frac{dT_b}{d\theta} \quad (17)$$

Substituting eqn. (16) into eqn. (17) thus yields

$$\frac{dT_b}{d\theta} = \frac{I}{m_b c_{p,b}} \left( -\frac{dQ_b}{d\theta} + \frac{V_b dP}{d\theta} + (h_u - h_b) \frac{dm_{x1}}{d\theta} \right) \quad (18)$$

(iii) Burned zone 2

By using the same procedure adopted in the analysis of the burned zone, the energy equation can be rearranged to obtain burned zone 2 temperature derivative as below.

$$\frac{d(m_{un} u_{un})}{d\theta} = \frac{dQ_{un}}{d\theta} - p \frac{dV_{un}}{d\theta} + \frac{h_u dm_{ub}}{d\theta} + \frac{h_b dm_{bu}}{d\theta} + \frac{h_{un} dm_{lun}}{d\theta} \quad (20)$$

mass in transitory zone ( $m_{un}$ )= mass infiltration from the unburned zone ( $m_{ub}$ ) + mass infiltration from burned zone ( $m_{bu}$ )

With  $\left[ \frac{dm_{un}}{d\theta} = \frac{dm_{ub}}{d\theta} + \frac{dm_{bu}}{d\theta} + \frac{dm_{b1}}{d\theta} + \frac{dm_{b2}}{d\theta} - \frac{dm_{lun}}{d\theta} \right]$ , where  $m_{lun}$  is mass leakage from the transitory zone to the crankcase, Eqn. (20) is re-expressed as

$$m_{un} c_{v,un} \frac{dT_{un}}{d\theta} = -\frac{dQ_{un}}{d\theta} - \frac{p dV_{un}}{d\theta} - u_{un} \frac{dm_{ub}}{d\theta} - u_{un} \frac{dm_{bu}}{d\theta} +$$

$$\frac{d(m_u u_u)}{d\theta} = -\frac{dQ_u}{d\theta} - p \frac{dV_u}{d\theta} - h_u \frac{dm_x}{d\theta} - \frac{h_u dm_{lu}}{d\theta} - \frac{h_u dm_{ub}}{d\theta} \quad (9)$$

$$\frac{dm_x}{d\theta} - \text{mass burning rate (MBR)}$$

$m_u$  - mass unburned

$m_{b2}$  - mass infiltration from burned zone

$m_{lu}$  - mass leakage from unburned zone to the crankcase

Eqn. (9) can be re-expressed as

$$\frac{d(m_u u_u)}{d\theta} = m_u \frac{du_u}{d\theta} + u_u \frac{dm_u}{d\theta} = m_u \left[ \frac{du_u}{dT_u} \cdot \frac{dT_u}{d\theta} \right] + \frac{u_u dm_u}{d\theta} \quad (10)$$

By noting that

$$\frac{dm_u}{d\theta} = -\frac{dm_{x1}}{d\theta} - \frac{dm_{lu}}{d\theta} - \frac{dm_{ub}}{d\theta} - \frac{dm_{x2}}{d\theta} \quad (11)$$

Eqn. (10) becomes

$$\frac{d(m_b u_b)}{d\theta} = \frac{dQ_b}{d\theta} - p \frac{dV_b}{d\theta} + h_u \frac{dm_x}{d\theta} - h_b \frac{dm_{lb}}{d\theta} - h_b \frac{dm_{bu}}{d\theta} \quad (15)$$

Mass of gas in burned zone ( $m_b$ ) = mass fraction burned ( $m_{x1}$ ) + mass infiltration from burned zone ( $m_{bu}$ )

With  $\left[ \frac{dm_b}{d\theta} = \frac{dm_{x1}}{d\theta} - \frac{dm_{lb}}{d\theta} - \frac{dm_{bu}}{d\theta} \right]$ , where  $m_{lb}$  is mass leakage from the burned zone to the crankcase, equation 15 is re-expressed as

Similarly, temperature for infiltration from burned zone to burned zone 2 is written as,

$$\frac{dT_{b2}}{d\theta} = \frac{I}{m_{b2} c_{p,b2}} \left( -\frac{dQ_{b2}}{d\theta} + \frac{V_{b2} dP}{d\theta} + (h_u - h_{b2}) \frac{dm_{x2}}{d\theta} \right) \quad (19)$$

(iv) Transitory ( obtained from infiltration of burned and unburned gases) zone.

In the same vein the energy equation for the transitory zone is written as

$$h_u \frac{dm_{ub}}{d\theta} + h_b \frac{dm_{bu}}{d\theta} \quad (21)$$

The ideal gas equation for this zone is expressed as

$$p \frac{dV_{un}}{d\theta} + V_{un} \frac{dP}{d\theta} = R_{un} T_{un} \frac{dm_{un}}{d\theta} + m_{un} R_{un} \frac{dT_{un}}{d\theta} \quad (22)$$

Substituting eqn. (21) into eqn. (22) thus yields

$$\frac{dT_{un}}{d\theta} = \frac{I}{m_{un}c_{p,un}} \left( -\frac{dQ_{un}}{d\theta} + V_{un} \frac{dP}{d\theta} + (h_u - h_{un}) \frac{dm_{ub}}{d\theta} + (h_b - h_{un}) \frac{dm_{bu}}{d\theta} \right) \tag{23}$$

When the overall energy equation is considered then the pressure derivative is obtained as

$$\frac{dP}{d\theta} = \left\{ \frac{V_u}{T_u} \frac{dT_u}{d\theta} + \frac{V_b}{T_b} \frac{dT_b}{d\theta} + \frac{V_{b2}}{T_{b2}} \frac{dT_{b2}}{d\theta} + \frac{V_{un}}{T_{un}} \frac{dT_{un}}{d\theta} + (v_b - v_u) \frac{dm_{x1}}{d\theta} + (v_{b2} - v_u) \frac{dm_{x2}}{d\theta} + (v_{un} - v_u) \frac{dm_{ub}}{d\theta} + (v_{un} - v_b) \frac{dm_{bu}}{d\theta} + (v_{un} - v_{b2}) \frac{dm_{b2}}{d\theta} - v_u \frac{dm_{lu}}{d\theta} - v_b \frac{dm_{lb}}{d\theta} - v_{b2} \frac{dm_{lb2}}{d\theta} - v_{un} \frac{dm_{lun}}{d\theta} - \frac{dV}{d\theta} \right\} / \frac{V}{P} \tag{24}$$

**2.2. Implementation of the Developed Model**

The developed model was used to obtain the temperature and pressure history in a combustion chamber. The five major ordinary differential equations (ODEs), namely equations 14, 18, 19, 23 and 24 for  $T_u, T_{b1}, T_{b2}, T_{un}$  and  $P$  were well arranged in a MATLAB program and used to describe the rates of change of the parameters with respect to crank angle. Equations 4-14 were used during compression and expansion phases while equations 15-23 were used during the combustion phase. Sub models including engine geometry, fuel data and air data inherent in the two-zone model as reported in [9] were used in analyzing the present four-zone model. By integrating these ODEs using the inbuilt function ODE45. m in MATLAB as solver, engine performance from start of compression to end of expansion is modeled and analysed.

The model was applied to an internal combustion engine with operating and combustion parameters details as contained in Table 1. In the analysis, consideration was given mass fraction ratio and mass infiltration into the transitory zone, such that predetermined optimal result were analysed for present case3.

In the analysis the mass infiltration into the transitory zone is such that:

$$m_{ub} = CC1 * m_u \text{ and } m_{bu} = CC2 * m_b$$

where CC1, fractional burnt leakage and CC2, fractional unburnt leakage are arbitrary constants.

Burned zone1 and burned zone2 have different mass burning rates, (x1, x2) using varying heat transfer coefficients, ( $h_{400}, h_{450}, h_{500}$ ),

*Table 1. Engine Specifications for 2Zone model.*

Parameter	Value
Number of cylinder	4
Bore	0.1
Stroke	0.08
Compression ratio	10

*Table 3. Four-zone model values in comparison to optimal mass infiltration and mass fraction burn values.*

CC1	CC2	Pe	IMEP	$\eta$	CO2	H2O	N2	O2	CO	H2	H	O	OH	NO
0.00025	0.005	4.4657e+06	9.4688e+05	0.3865	0.0983	0.1194	0.7428	0.0395	0.0000	0.0000	0.0000	0.0000	0.0000	0.0001
X1	X2													
0.6	0.4	4.4782e+06	9.5042e+05	0.3880	0.0983	0.1194	0.7428	0.0395	0.0000	0.0000	0.0000	0.0000	0.0000	0.0001
CASE3 VALUES (4zone model)		4.4969e+06	9.4545e+05	0.3859	0.0983	0.1194	0.7428	0.0395	0.0000	0.0000	0.0000	0.0000	0.0000	0.0001

Parameter	Value
Equivalence ratio	0.8
Burn duration angle	60
Start of burning	-35
Engine speed	1500RPM

[9]

*Table 2. Engine Specifications for model evaluation.*

Parameter	Value
Number of cylinder	4
Bore	0.085
Stroke	0.078
Compression ratio	11
Displacement	1.018
Start of Combustion	21
End of Combustion	30
Engine speed	2000RPM

[11]

The performance parameters such as thermal efficiency ( $\eta$ ) and indicated mean effective pressure (IMEP) as well as the emission fraction obtained using the developed model were compared with the standard two-zone model and both were compared to an experimental result from a four stroke, SI engine (EF7) running with CNG fuel, [11], engine geometry are shown in Table 2. Results for IMEP,  $\eta$ , Pressure and emission characteristics were presented graphically in figures 2 to 5 and tables 3and 4.

**3. Results and Discussion**

The conclusive results from both analyses of mass infiltration and mass fraction burn are used for analyzing the 4zone model. Extensive explanation of the derived mass infiltration and mass fraction analysis can be found in [10, 12] respectively. Table 3 shows the four-zone model values in comparison to optimal mass infiltration and mass fraction burn values.

Table 4. Comparison of 4zone results from 2zone model, 3zone model and experimental results.

	2zone model	Experimental values	3zone model using $x_1=0.6$ and $x_2=0.4$	3zone model using $CC_1=0.00025$ and $CC_2=0.005$	4zone Model using $x_1=0.6$ and $x_2=0.4$ ( $h_{400}$ ), $CC_1=0.00025$ and $CC_2=0.005$
Pe	4.4358e+06	4.640e+06	4.5185e+06	4.4657e+06	4.4969e+06
Tbe	2305K	2800K	23157K	2.3241e+03	2.315e+03
IMEP	9.1635e+05		9.4847e+05	9.4688e+05	9.4545e+05
$\eta$	0.3741	-	0.3872	0.3865	0.3859

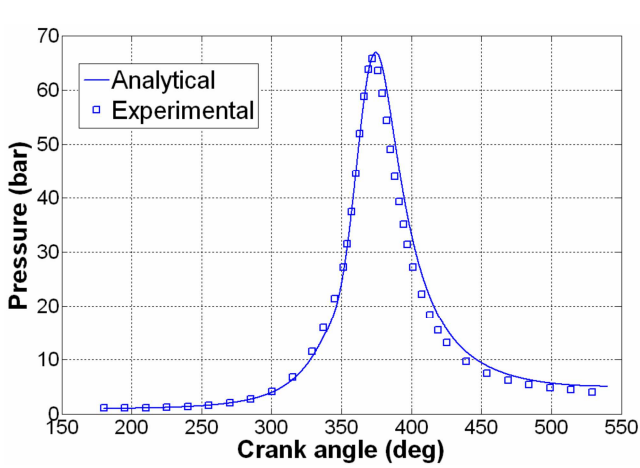


Figure 2. Pressure history for experimental Model (Omid, 2012).

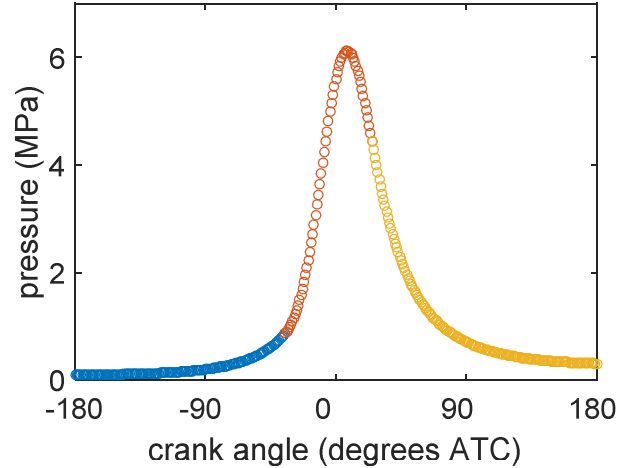


Figure 3. Pressure history for 2zone Model.

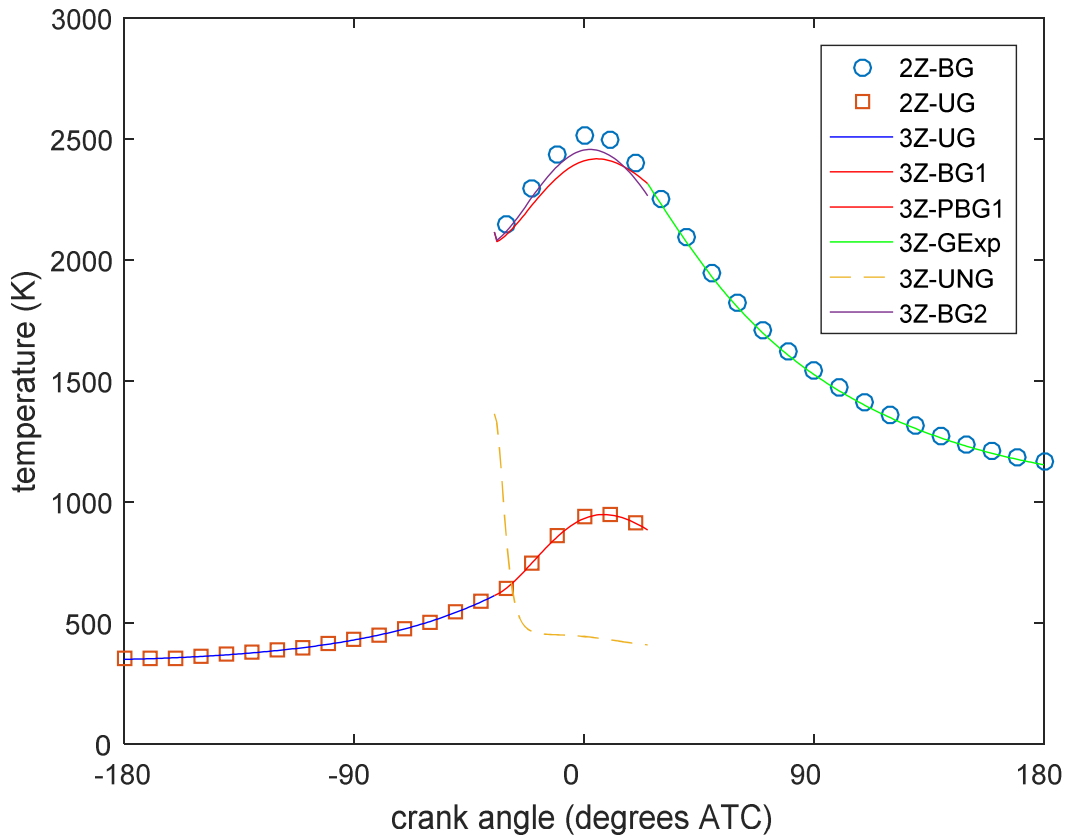


Figure 4. Temperature history for 4zone Model using  $x_1=0.6$  and  $x_2=0.4$  ( $h_{400}$ ),  $CC_1=0.00025$  and  $CC_2=0.005$ .

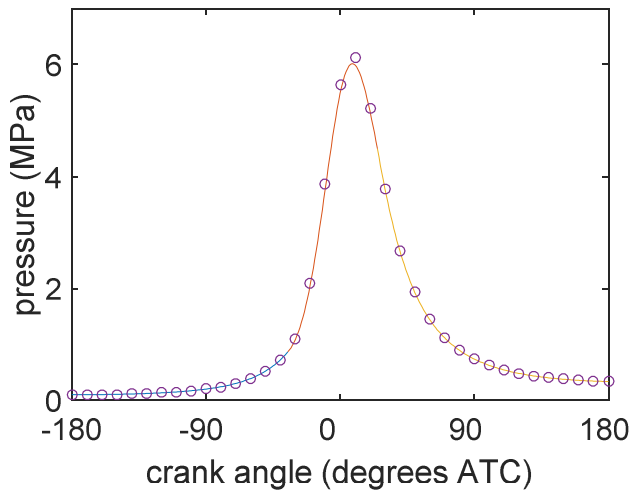


Figure 5. Pressure history for 4zone Model using  $x_1 = 0.6$  and  $x_2 = 0.4$  ( $h_{400}$ ),  $CCI = 0.00025$  and  $CC2 = 0.005$ .

Maximum peak pressure of 4.4969MPa was observed using 4zone model. It could be inferred that the 4zone model is better at predicting pressure history, see table 3. Highest values of IMEP and  $\eta$  were observed using the 3zone segmented model ( $x_1 = 0.6$ ,  $x_2 = 0.4$ ). It was observed that the 4zone model temperature history is identical to the 2zone model, see figure 4.

A comparison of the experimental, 2zone, 3 zone and 4zone models results are presented in Table 4. Higher peak pressure values were observed for 4zone model which was 3.1% closer to experimental value as compared to the 2zone model which was 4.4% closer. This could imply an improved engine work estimate using the 4zone model. Higher peak temperature values were observed for 4zone model which was 17% closer to experimental value as compared to the 2zone model which was 18% closer. This could imply a better temperature distribution achieved using the 4zone model. Thermal efficiency and IMEP values were observed appreciably higher for the 4-zone model as compared with the 2-zone model.

## 4. Conclusion

A 4-zone model for prediction of performance of an internal combustion engine has been developed. The results obtained established that incorporating a new zone by dividing the burned zone into segments, comprising variable mass fraction burned coupled with an additional zone comprising mixture of unburned and burnt gases at a determined optimal range, the 4-zone model gave a better estimate of an internal combustion engine performance characteristics.

## Nomenclature

W: work (J)  
 P: pressure (Pa)  
 $c_v$ : specific heat at constant volume (J/kg/K)

$c_p$ : specific heat at constant pressure (J/kg/K)  
 h: enthalpy (J/kg)  
 m: mass (kg)  
 Q: heat transfer rate across the system boundary into the system (J)  
 R: specific gas constant (J/kgK)  
 T: temperature (K)  
 u : internal energy of material contained in the system (J/kgK)  
 v: volume ( $m^3$ )  
 $\theta$ : crank angle ( $^\circ CA$ )  
 $m_x$ : mass fraction burned

## Subscripts

$b_1$ : burned zone 1  
 u: unburned  
 $b_2$ : burned zone 2  
 un: unburned burned  
 $v_1$ : volume of fuel A  
 $v_2$ : volume of fuel B  
 l: leakage/blowby  
 $mx_1$ : mass burn rate for fuel A  
 $mx_2$ : mass burn rate for fuel B

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