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Two Zone Combustion Model for SI Engines to Study the Variations in Cylinder Pressures and Temperatures between Adiabatic Combustion and Eichelberg's Correlation

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Abstract **- Computer simulation of SI engines is a useful tool to predict engine performance due to changes in input variables and also study the effect of phenomena like heat transfer to the engine performance. The paper describes the building of a thermodynamic two zone combustion model for SI engines to observe the theoretical deviation in cylinder pressure and temperature between a case of adiabatic combustion and a case using Eichelberg's heat transfer correlation. The two zone model developed in MATLAB is a good platform to analyse such results as it can depict changes in peak burned gas temperature that single zone models can't and saves on the long computation time required in multi zone models. Existing two zone models do not study the effect of heat transfer on the engine's output. The work aims to find the deviation of engine cylinder pressure and temperature over a range of engine speeds and over the entire crank angle duration. The simulation shows that the pressure deviation was small compared to the temperature deviation, especially at higher engine speeds, where the peak pressure variations became negligible. The combustion model was verified using experimental data from referenced works.**
Keywords: ***Engine-Model-Combustion***

I. INTRODUCTION

Computer simulation of engines is an important tool in analyzing the effect of input variables like intake pressure, valve timings and spark advance. It helps in optimizing an engine design for particular applications. It also reduces the need for expensive engine tests and provides a cost effective method for engine study.

The simulation of combustion in SI engines can be done using zero dimensional or multi-dimensional combustion models. Zero dimensional models are purely thermodynamic models, they do not model the flow of the mixture inside the combustion chamber. Zero dimensional models can be further subdivided into single zone, two zone and multi zone models. Single zone models treat the entire combustion chamber as a single zone in which heat is added or subtracted. The zone has uniform properties throughout and has the same values of combustion pressure and temperature at all points. Ferguson's [2] original work had a FORTRAN code for a combustion model which was widely used. However the model was a single zone model and its accuracy was limited.

Two zone models divide the combustion chamber into burned gas mixtures and unburned gas mixtures with the flame front dividing the zones. The mixture which is further away from the spark plug is first compressed due to the increasing pressure generated by the advancing flame front and burning mixture and then burned, whereas the mixture closer to the spark plug is first burned and then compressed. This leads to a gradient of properties inside the combustion chamber. Multizone models hence, divide the combustion chamber into multiple zones, each with homogenous properties.

However with increasing complexity, the computation time required for the model also increases.

Heat transfer is a critical aspect in internal combustion engines. All thermodynamic processes within an IC engine are never truly adiabatic and heat is always rejected to the engine coolant through the combustion chamber walls to keep the engine temperatures in safe operating conditions. The two zone combustion chamber incorporates an Eichelberg heat transfer model to calculate the heat transfer coefficient to the walls of the combustion chamber.

Current simulations of Eichelberg's heat transfer correlation do not study the fundamental deviation of cylinder pressures and temperatures from adiabatic combustion due to it. This paper aims to study the behavior of Eichelberg's model and predicts the changes in engine output due to enabling heat transfer to the cylinder wall.

II. ASSUMPTIONS TAKEN

1. The model assumes that complete burning of the fuel takes place. That is, in cases of the fuel air equivalence ratio being less than 1, all fuel is converted to carbon dioxide and water vapor.
2. Unthrottled operation is assumed.
3. No valve overlap is allowed in the model.
4. The combustion chamber is assumed as disc shaped and the surface area of the burned zone with respect to the flame front radius is taken according to this assumption.
5. Eichelberg's heat transfer model is taken for the computation of heat transfer coefficient.
6. The specific gas constant for each gas is taken as a constant and does not vary with temperature or pressure

(dR = 0)

7. The flame front is assumed be perfectly spherical with the spark plug as the centre.
8. The motion of the charge or the gases inside the cylinder are not accounted for due to it being a zero dimensional model.

III. SIMULATION MODEL

The properties of each zone are calculated for each time step in the model based on the properties in the previous time step. Each time step corresponds to 0.1 degree of crank angle rotation. The model is sub divided into a number of sub-routines to simulate a particular process within the combustion chamber. The sub routines are intake, compression, combustion, expansion and exhaust.

A. Governing Equations For Combustion

The governing equations for the two zones can be written as –

$$dV_u + dV_b = dV \tag{1}$$

$$V_u dP + PdV_u - m_u R_u dT_u = R_u T_u dm_{ub} \tag{2}$$

$$PdV_u + m_u c_{v,u} dT_u = -dQ_u + (h_{ub} - u_u) dm_{ub} \tag{3}$$

$$V_b dP + PdV_b - m_b R_b dT_b = R_b T_b dm_{bu} \tag{4}$$

$$PdV_b + m_b c_{v,b} dT_b = -dQ_b + (h_{bu} - u_b) dm_{bu} \tag{5}$$

Where dV, dP and dT denote the changes in volume, pressure and temperature respectively for the zones. These values are calculated for each time step and are then added to the values at the previous iteration.

According to Nilsson [5], the gas is assumed to undergo combustion instantly as it leaves the unburned zone and enters the burned zone. This means that:

$$h_{ub} = h_{bu} = h_u \tag{6}$$

The value of dm is calculated using the cosine burn law for each 0.1 degree crank rotation. The fundamental formulation for the equations are taken from Nilsson[5].

The equations can be arranged in the form of

$$AX = B \tag{7}$$

Where

$$A = \begin{bmatrix} 0 & 1 & 0 & 1 & 0 \\ V_u & P & -m_u R_u & 0 & 0 \\ 0 & P & m_u c_{v,u} & 0 & 0 \\ V_b & 0 & 0 & P & -m_b R_b \\ 0 & 0 & 0 & P & m_b c_{v,b} \end{bmatrix} \tag{8}$$

$$X = \begin{bmatrix} dP \\ dV_u \\ dT_u \\ dV_b \\ dT_b \end{bmatrix} \tag{9}$$

$$B = \begin{bmatrix} dV \\ R_u T_u dm \\ -dQ_u + R_u T_u dm \\ R_b T_b dm \\ -dQb + (h_u - h_b + R_b T_b) dm \end{bmatrix} \tag{10}$$

This makes the solving easier as MATLAB can simply find the inverse of matrix A and multiply it to matrix B to get X. Adaptive scaling of the matrices is done to keep the condition number of the matrix within acceptable limits.

MATLAB provides a suitable platform for writing the code for such a simulation model, as pointed out by Buttsworth[8] in his work where he compares his code written in MATLAB to Ferguson's[2] code written in FORTRAN.

Properties of both unburned and burned gases were taken from Heywood[1]. These properties are approximations and reasonably conform to those according to JANAF[10] tables and Newhall's[9] work. The mass fraction burn rate was taken as a cosine function of the angle of combustion duration. The behavior of the mass burnt fraction with increasing crank angle was according to established norms.

An approach in which an empirical relation for flame front velocity would be used to calculate mass burnt fraction was also considered, similar to that done by Hosseini[7], this however did not give good agreement with combustion duration values for crank angle.

$$n = \frac{1}{2}\left[1 - \cos\left[\frac{\theta - \theta_i}{\Delta\theta_c}\pi\right]\right] \tag{11}$$

Where, n is the mass burned fraction, θ is the crank angle and θ_i is the crank angle at the start of combustion. $\Delta\theta_c$ is the duration of the combustion process and is taken according to Taylor's[4] formulation as

$$\Delta\theta_c = 40 + 5\left(\frac{N}{600} - 1\right) + 166(\emptyset - 1.1)^2 \tag{12}$$

At each time step, the properties dV, dP and dT are calculated and the values are updated for the next time step.

B. Intake and Exhaust

The intake and exhaust simulation has been done similar to Ganesan's[3] single zone combustion model.

For simulating the flow across a restricted area, the intake and exhaust flow must be considered in two regimes – subsonic and supersonic. The ratio of upstream pressure to downstream pressure ($^{P}/_{p_0}$) must be under a critical value for subsonic flows. In supersonic flows the flow across the valve becomes choked, and the mass flow rate of the gas across the valve becomes constant, independent of the pressure ratio existing across the valve.

$$Critical\ Pressure\ Ratio = \left(\frac{\gamma + 1}{2}\right)^{\frac{\gamma}{\gamma - 1}} \tag{13}$$

Where γ = specific heat ratio of the gas or mixture

The mass flow rate across a restriction or the valve in case of subsonic flow is

$$\frac{dM}{dt} = Ap_0\sqrt{\frac{2\gamma}{RT(\gamma-1)}\left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}}\left[\left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}}-1\right]} \tag{14}$$

And in case of supersonic flow is

$$\frac{dM}{dt} = AP\sqrt{\frac{\gamma}{RT}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{15}$$

C. Heat Transfer Model

Heat transfer within the internal combustion engine is an important aspect of the combustion process. It maintains the cylinder head, cylinder wall and the piston in safe operating conditions and keeps the temperatures lower than their melting point.

The equation for heat transfer to the cylinder walls is the standard equation for heat transfer due to convection.

$$Q = h_c A \Delta T \tag{16}$$

Where h_c is the heat transfer coefficient, A is the cylinder wall area and ΔT is the difference in temperature between the cylinder wall and the gas.

The problem however lies in determining the value of heat transfer coefficient of the gases to the cylinder walls. Various models and empirical formulae exist to calculate the value of the heat transfer coefficient. According to Luonici[6], Eichelberg's correlation has an acceptable accuracy, uses relatively low computation times and requires no tuning. The relation is given by

$$h_c = 2.1(C_m)^{1/3}(PT)^{1/2} \tag{17}$$

The area of the zones in contact with the cylinder walls was determinedby using the burned gas volume to determine the radius of the flame front from the spark plug. And the area was then determined from the enflamed region radius.

IV. RESULTS

The simulation was carried out under the following engine conditions. The data was taken similar to the engine specifications used by Ganesan[3].

Bore	79.4 mm
Stroke	111.2 mm
V_{disp}	550.8 cc
Connecting Rod Length	233.4 mm
Compression Ratio	7.4
Operation	Full Throttle
Intake manifold Pressure	1 atm
Intake Manifold Temperature	300 K
Air Fuel Equivalence Ratio	1.0
Speed	4000 RPM
Intake Valve Opening/ Intake Valve Closing	0 degree BTC/ 30 degrees ABC
Exhaust Valve Opening/ Exhaust Valve Closing	30 degree BBC/ 0 degree ATC

The simulation results can be seen in Figure 1 and Figure 2. Comparing the cylinder pressure data to Ganesan's[3] model, the simulation done has a peak pressure value closer to experimental pressure data than Ganesan's one zone model.

Figure 1 shows the drop in pressure due to activating Eichelberg's heat transfer correlation. The pressure drop is not significant during the earlier phase of combustion and the pressure difference is only slightly larger by the end of the expansion stroke, late in the expansion stroke. The reduction in peak pressure is only1.9% whereas the maiximum change in the pressure is 1.165 bar, which is 4.6%.

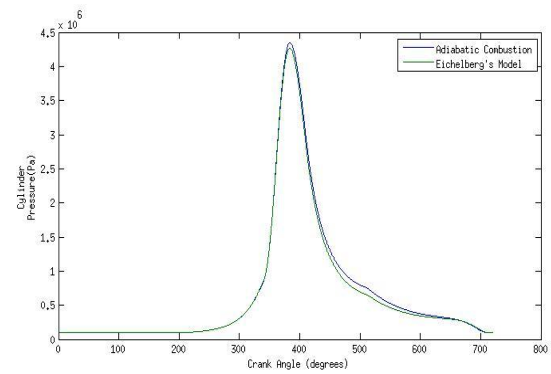

Figure 1: Pressure Comparison between Adiabatic combustion and Eichelberg's Heat Transfer Model

Figure 2 shows the variation of burned gas temperatures in the two zone model. The temperature difference in the zone is significant during the expansion stroke. The maximum drop in temperature in this case was 266.08 K which is a 13.37% change in cylinder temperature.

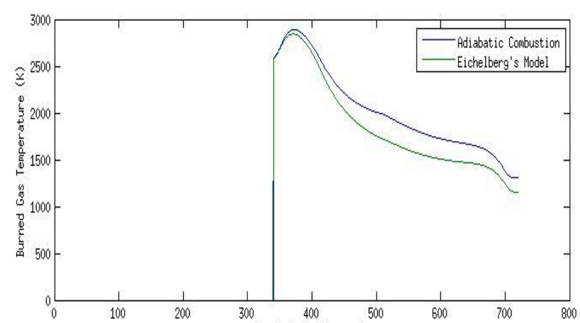

Figure 2: Burned gas temperature comparison between adiabatic combustion and Eichelberg's heat transfer model

Figure 3 shows the difference in peak cylinder pressures over a range of engine speeds. The difference in pressures is greater at lower speeds, than at higher speeds, due to longer times that the gas has to reside in the cylinder, hence convecting away more heat to the cylinder walls.

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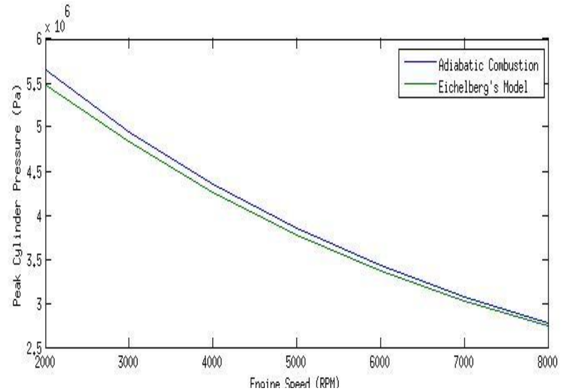

Figure 3: Peak cylinder pressure comparison between adiabatic combustion and Eichelberg's heat transfer model over a range of 2000 rpm - 8000 rpm

Figure 4 shows the variation in peak temperatures over the same engine speed range.

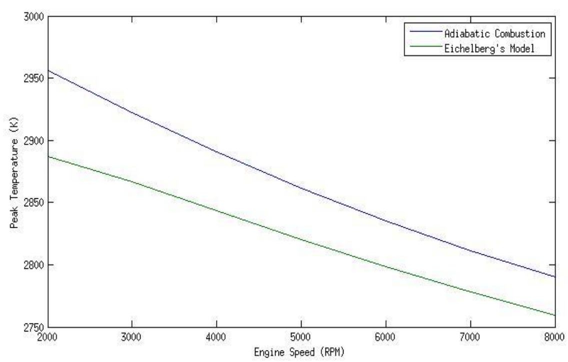

Figure 4:Peak temperature comparison between adiabatic combustion and Eichelberg's heat transfer model over a range of 2000rpm-8000rpm

V. CONCLUSION

The two zone model was successfully implemented and the integration of the Eichelberg's heat transfer model was done. The variations in cylinder pressure and burned gas temperatures can be seen in the simulation results.

The effect on temperature in the later stages of the expansion process and the exhaust stroke is significant and the burned gas temperature decreases by greater than 250 K. The deviation in cylinder pressure is not as significant during the engine cycle.

The variation in peak temperatures and pressures with changing engine speed showed a decreasing change in pressure. The peak pressure change was almost negligible at 8000 RPM. Peak cylinder temperatures still show a large variation even at higher engine speeds, though they show a decreasing trend.

VI. NOMENCLATURE

V - Volume (m^3)
P – Pressure (Pa)
R – Specific Gas Constant ($JK^{-1}g^{-1}$)
T – Temperature (K)
M – Mass (g)
c_v – Specific Heat Capacity at Constant Volume ($JK^{-1}g^{-1}$)
Q – Heat Transferred to the Cylinder Walls (J)
h – Specific Enthalpy (Jg^{-1})
u – Specific Internal Energy (Jg^{-1})
n – Mass Burnt Fraction
N – Engine Speed (RPM)
k – Ratio of specific heats ($^{c_p}/_{c_v}$)
C_m – Mean Piston Speed ($m\ s^{-1}$)
A – Valve Curtain Area (m^2)

Subscripts Used
u –unburntmixture property
b –burntmixture property
ub – property of mixture going from unburnt to burnt zone
bu – property of mixture going from burnt to unburnt zone

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