119 A New Approach to Simulation of Combustion in Gasoline Engines

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Combustion modeling is a very important method in designing internal combustion engine. In this paper, a two-zone model and also a multi-zone one with diagnostic and predictive functions are proposed for combustion in a gasoline engine. These models have been used to predict the performances and their changing with variable parameters such as Ignition Advance Angle and Coefficient of the Residual Exhaust Gas respectively, in a BJ492Q gasoline engine. All the results of these calculations have been compared with experimental measurements. The results show that the proposed models are very valid and effective.

Keywords: Gasoline Engine, Combustion, Simulation, Multi-zone Model

Nomenclature

A_f Area of flame front
c_p Specific heat at constant pressure
c_v Specific heat at constant volume
F_r Coefficient of the residual exhaust gas
H Enthalpy of mixture
m Mass of the mixture in combustion chamber
m_z Mass of the mixture in each small zone in the multi-zone model
n Number of the small zones in the unburned zone divided by the equipartition of mass; Rotative velocity of engines
n_A Serial number of the preparing zones
p Pressure in cylinder
p_{max} Maximum of indicated pressure in cylinder
P_e Effective power
Q Heat loss of mixture
r Radius of curvature of the flame front
R Gas constant
T Average temperature of mixture
u' Intensity of the turbulence in unburned zone
U Internal energy of mixture
v_T Velocity of turbulent flame
V Volume of combustion chamber
W Indicated work

Greek Symbols

ε Ratio of the turbulent to the laminar burning velocity
Φ Equivalence ratio
φ Crank angle
ρ Density of mixture

Subscripts

b Burned zone
i Burned small zone in the multi-zone model (i=1,2,⋯,n)
u Unburned zone
x Preparing zone (in this zone, mixture is burning)

1. Introduction

In this paper, a new two-zone and a new multi-zone model are proposed for combustion in a gasoline engine. These models can be used for analyzing combustion chambers with different structures, predicting their performances and optimizing their configurations. Assuming that a flame front propagates in spherical waves [1], the flame front divides a combustion chamber into unburned and burned zones. Each of the two zones is assumed to be in thermodynamic equilibrium. The rate of burning fuel is taken to be the product of the speed of a turbulent flame, the area of the flame front and the density of the unburned mixture. Consequently, when calculating the rate of combustion, the effects of turbulence are included, with the geometry of the combustion chamber. The empirical ratio of the turbulent to the laminar burning velocity can be regressed. These results in combustion being simulated are better than those by a single-zone model, where the burning rate is given in advance.

2. Fundamental Equations for Combustion Models

Figure 1[2] depicts a flame front moving in a spherical shape, and radius of curvature r. The combustion chamber in Fig.1 is divided into an unburned zone and a burned zone, whose volumes are V_u and V_b, respectively, average temperatures T_u and T_b, respectively, and the common pressure is p. The upper boundary of the combustion chamber is the irregularly shaped cylinder head and its lower boundary is the piston head. There is no heat transfer between the two zones. The unburned mixture contains air, fuel vapor and some residual exhaust gas; the burned mixture consists of combustion products, whose concentrations could be calculated assuming chemical equilibrium, of course, except NO_x.

Based on above-mentioned assumptions, in terms of First Law of thermodynamics, the conservation equations of energy can be deduced, respectively, for the burned and unburned zones, as follows:

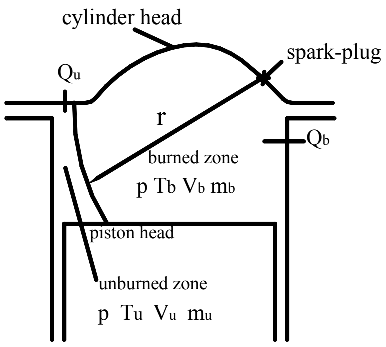

Fig. 1 Schematic diagram of the two-zone combustion model

$$\dot{\overline{m_u U_u}} = H_u \dot{m}_u + \dot{Q}_u - p\dot{V}_u \quad (1)$$

$$\dot{\overline{m_b U_b}} = H_u \dot{m}_b + \dot{Q}_b - p\dot{V}_b \quad (2)$$

In this paper, the superscript dots means differentiating with respect to the crank angle φ.

In unburned and burned zones, the state equations of perfect gas are, respectively:

$$pV_u = m_u R_u T_u \quad (3)$$

$$pV_b = m_b R_b T_b \quad (4)$$

Assuming there is no leakage from the combustion chamber, the equations of mass and volume for this closed system are:

$$m = m_u + m_b = cons\tan t \quad (5)$$

$$V = V_u + V_b \quad (6)$$

The relationship of the rate of burning fuel, the velocity of the turbulent flame and the area of the flame front can be expressed as:

$$\dot{m}_b = \rho_u A_f \upsilon_T \quad (7)$$

Equations 1 ~ 7 are the fundamental equations for combustion models.

3. Combustion Models

3.1 Two-Zone Models

3.1.1 Diagnostic Model

By experimental measurements of an engine, a diagnostic model can be used to analyze the performances of the engine and their changes. In other words, the values of $\dot{m}_b$, T_u, T_b and V_b can be obtained as long as p and $\dot{p}$ are known. The model can be described as:

$$\dot{T}_u = \left(T_u \frac{\dot{p}}{p} + \frac{\dot{Q}_u}{m_u R_u}\right) / \left(1 + \frac{1}{R_u}\frac{\partial U_u}{\partial T_u}\right) \quad (8)$$

$$\dot{V}_b = V_u\left(\frac{\dot{m}_b}{\dot{m}_u} - \frac{\dot{T}_u}{T_u} + \frac{\dot{p}}{p}\right) + \dot{V} \quad (9)$$

$$\dot{m}_b = A / B \quad (10)$$

where

$$A = \left(m_u R_u \dot{T}_u - p\dot{V}\right)\left(1 + \frac{c_{vb}}{R_b}\right)$$

$$- \dot{p}\left(m_b \frac{\partial U_b}{\partial p} + V_u + c_{vb}\frac{V}{R_b}\right) + \dot{Q}_b$$

$$B = U_b - U_u + \frac{c_{vb}}{R_b}\left(R_u T_u - R_b T_b\right)$$

In equations 8~10, U, H, R and their partial derivatives can be solved by calculation programs which are specially made for calculating the characteristics of the working substance.

After the thermodynamic parameters such as T_u, T_b, m_b and $\dot{m}_b$ have been obtained, the area of flame front can be solved by a geometry model, then, the ratio of the turbulent to the laminar burning velocity can be obtained.

3.1.2 Predictive Model

Contrary to the diagnostic model, in predictive model, the relative movement of the flame front to the unburned mixture is given as a precondition. That is to say, $\dot{m}_b$ can be solved by the empirical ratio of the turbulent to the laminar burning velocity, then, some thermodynamic parameters and other performance parameters can be obtained. The model can be described concretely as:

The rate of pressure change

$$\dot{p} = D / E \tag{11}$$

where

$$D = \left(1 + \frac{c_{vb}}{R_b}\right) p\dot{V} + \left[\left(U_b - U_u\right) - c_{vb}\left(T_b - \frac{T_u R_u}{R_b}\right)\right] \cdot \dot{m}_b + \left(\frac{c_{vu}}{c_{pu}} - \frac{c_{vb}}{c_{pu}} \cdot \frac{R_u}{R_b}\right)\dot{Q}_u - \left(\dot{Q}_u + \dot{Q}_b\right)$$

$$E = -\left(V_u \cdot \frac{c_{vu}}{c_{pu}} - \frac{R_u c_{vb} V_u}{R_b c_{pu}} + \frac{V c_{vb}}{R_b} + m_b \cdot \frac{\partial U_b}{\partial p}\right)$$

The rate of temperature change in the unburned zone

$$\dot{T}_u = \frac{V_u \dot{p} + \dot{Q}_u}{m_u c_{p_u}} \tag{12}$$

The rate of temperature change in the burned zone

$$\dot{T}_b = [p\dot{V} - \left(R_b T_b - R_u T_u\right)\dot{m}_b - \frac{R_u}{c_{pu}}\left(V_u \dot{p} + \dot{Q}_u\right) + V\dot{p}]/(m_b R_b) \tag{13}$$

The pressure and temperature in a cylinder can be obtained as long as $\dot{m}_b$ is known, then, other parameters can be worked out and predictive calculation can be performed.

3.2 Multi-zone Model

One of the specialties of combustion in a gasoline engine is that the flame front propagates from layer to layer. Therefore, the mixture in the combustion chamber is inhomogeneous at different location and different time. In the two-zone model, the burned zone is assumed to be a whole and each calculation result is an average of it. To describe the characteristics of the combustion more realistically, the burned zone can be divided into some small zones. Assuming these small zones are adiabatic each other and there is no mixing, then, each small zone is a single-zone system and in a state of chemical equilibrium. The common pressure of the whole zone is still p.

Usually, there are three methods to divide the burned zone: the first considers an equipartition of mass, the second and third involve equipartitions of radius and crank angle, respectively. By comparison, the first method is simpler. By this method, the mass of the mixture in each small zone is constant and the mutual effects between them are considered by use of the change of their volumes [3]. In this paper, the calculation is carried out based on this method shown in Figure 2.

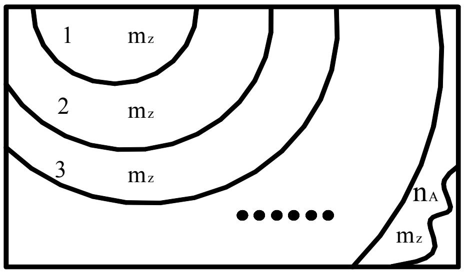

Fig. 2 Schematic diagram of the multi-zone model divided by the equipartition of mass

Supposing that the total mass of the mixture in a combustion chamber is m and it is divided into n parts, then, the mass in each small zone is $m_z = m/n$. Again, supposing that m_x is the mass of burned mixture in a preparing zone (in this zone, mixture is burning) and n_A is the serial number of the preparing zones. In combustion, a new preparing zone is formed, if

$$m_x \geqslant m_z \tag{14}$$

This process continues until

$$n_A = n \tag{15}$$

Initially, the temperature in a new preparing zone is equal to that in the zone previous to it. The excess mixture (m_x - m_z) in the previous zone should be calculated in the new preparing zone and the mass of NO_x is also brought into it in proportion to the excess mixture. The small burned zones are in the state of transferring heat to walls, powering piston and being compressed by the small zones formed subsequently.

3.2.1 Diagnostic Model

The rate of temperature change in burned small zone i

$$\dot{T}_i = \frac{V_i \dot{p} + \dot{Q}_i}{m_z\left(R_i + c_{vi}\right)}, i = 1,2,3,\cdots n \tag{16}$$

The rate of temperature change in the unburned zone is

$$\dot{T}_u = \frac{V_u \dot{p} + \dot{Q}_u}{m_u c_{pu}} \tag{17}$$

The rate of change in the mass of burned mixture in a preparing zone is

$$\dot{m}_x = A_x / B_x \tag{18}$$

where

$$A_x = \left(m_u R_u \dot{T}_u - p\dot{V}\right)\left(1+\frac{c_{vx}}{R_x}\right) - \dot{p}\left(m_x \frac{\partial U_x}{\partial p} + V_u + \frac{c_{vx}}{R_x} V\right) + \dot{Q}_x + p\sum \dot{V}_i + \frac{c_{vx}}{R_x}\sum m_z R_i \dot{T}_i$$

$$B_x = \left(U_x - U_u\right) + \frac{c_{vx}}{R_x}\left(R_u T_u - R_x T_x\right)$$

The rate of volume change in a preparing zone is

$$\dot{V}_x = V_u\left(\dot{p}/p + \dot{m}_x / m_u - \dot{T}_u / T_u\right) + \dot{V} - \sum \dot{V}_i \tag{19}$$

Equations 16～19 constitute the diagnostic model of the multi-zone model.

3.2.2 Predictive Model

The rate of pressure change is

$$\dot{p} = D_x / E_x \tag{20}$$

where

$$D_x = -\left\{\left(1+\frac{c_{vx}}{R_x}\right)p\dot{V} + \left[\left(U_x - U_u\right) - c_{vx}\left(T_x - \frac{R_u T_u}{R_x}\right)\dot{m}_x\right]\right.$$

$$\left. + \left(\frac{c_{vu}}{c_{pu}} - \frac{c_{vx} R_u}{c_{pu} R_x} - 1\right)\dot{Q}_u - \dot{Q}_x - \frac{c_{vx}}{R_x}\sum \frac{R_i}{R_i + c_{vi}}\dot{Q}_i - p\sum \dot{V}_i\right\}$$

$$E_x = V\frac{c_{vx}}{R_x} - \frac{c_{vx} R_u V_u}{R_x c_{pu}} + \frac{c_{vu} V_u}{c_{pu}} + m_x \frac{\partial U_x}{\partial p} + \frac{c_{vx}}{R_x}\sum \frac{R_i V_i}{R_i + c_{vi}}$$

The rate of temperature change in the unburned zone is

$$\dot{T}_u = \left(V_u \dot{p} + \dot{Q}_u\right) / \left(m_u c_{pu}\right) \tag{21}$$

The rate of temperature change in burned small zone i

$$\dot{T}_i = \left(V_i \dot{p} + \dot{Q}_i\right) / \left(m_z\left(R_i + c_{vi}\right)\right) \tag{22}$$

The rate of temperature change in a preparing zone is

$$\dot{T}_x = \frac{p}{m_x R_x}\left[\dot{V} + V\frac{\dot{p}}{p} - \left(\frac{R_x T_x}{p} - \frac{R_u T_u}{p}\right)\dot{m}_x - \frac{V_u R_u \dot{p}}{p c_{pu}} - \frac{R_u \dot{Q}_u}{p c_{pu}} - \frac{m_z}{p}\sum R_i \dot{T}_i\right] \tag{23}$$

Equations 20～23 constitute the predictive model of the multi-zone model. Combined with sub-models such as the model of the characteristics of the mixture and the geometry model, the predictive calculation can be carried out by a fourth order Runge-Kutta method and.

4. Validation

In order to verify the effectiveness of the combustion models, the indicator diagram and the characteristic parameters at full throttle in a BJ492Q gasoline engine were measured, and the corresponding data were calculated. Figure 3 gives the comparison of these measurements with the calculation results.

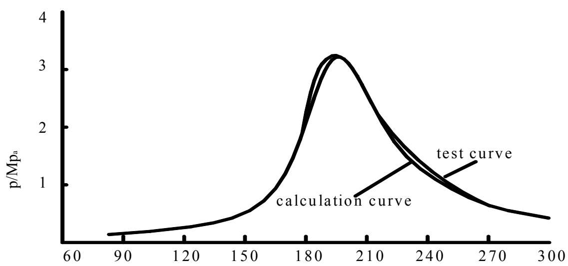

Fig. 3 Comparison of the experimental measurements and the calculation results of the indictor diagram at 2500 RPM, 24.9kW, with the coefficient of the residual exhaust gas Fr=0.94 in a BJ492Q gasoline engine

It is known from Figure 3 that the relative error of the pressure in cylinder is less than 5%, and that of indicated work is less than 4% during combustion. Obviously, the computations have been improved due to that the effects of turbulence on combustion were considered. The case of error comes from the assumptions made in the models. Another reason is the lack of precise experimental measurements of the coefficient of heat transfer, the coefficient of residual exhaust and the transient temperature of chamber walls.

5. Prediction of Performances Based on the Two-Zone Model

5.1 Effects of the Change of Ignition Advance Angle on Performance Parameters

The experimental gasoline engine was a production BJ492Q with operating conditions 2500 r/min, 29.4 kW. In calculations, a single parameter is changed at a time. The effects of the changes on combustion can be predicted, and some important factors relative to the saving of energy and pollution control in gasoline engines can be found.

Figure 4 gives the calculation results of the pressure, temperature, velocity of turbulent flame and the concentration of NO_x with the change of ignition advance angle. Obviously, with the reduction of ignition advance angle, the curves of indicator diagram moves rightwards, the increase of the pressure slows down, the maximums of the temperature and velocity of the turbulent flame go down and the amount of NO_x lessens.

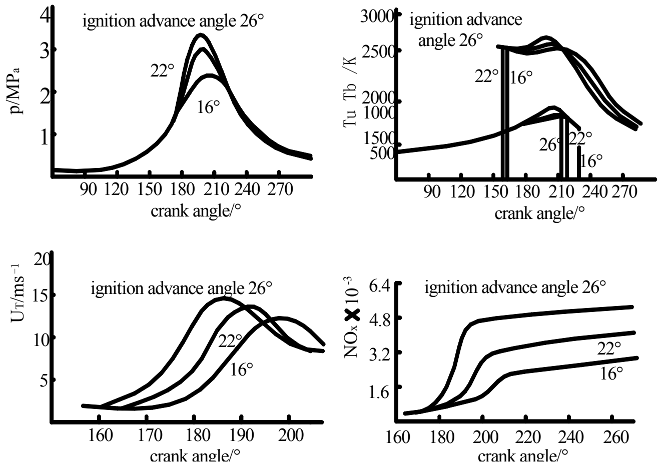

Fig. 4 Effects of the change of ignition advance angle on the performance parameters at 2500 RPM, 29.4kW in a BJ492Q gasoline engine

5.2 Effects of the Change of the Coefficient of the Residual Exhaust Gas on Performance Parameters

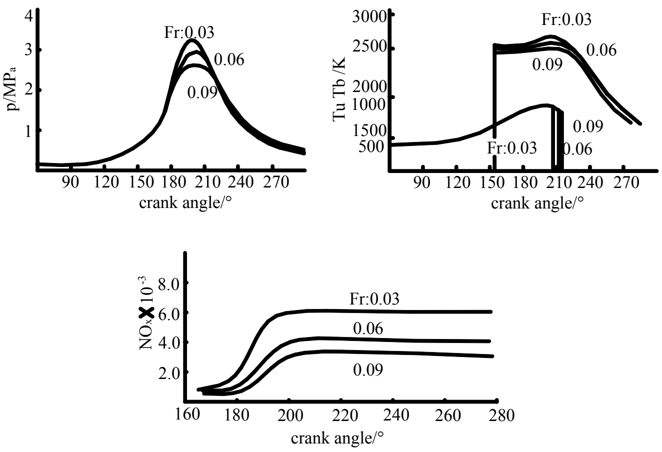

Fig. 5 Effects of the change of the coefficient of the residual exhaust gas on the performance parameters at 2500 RPM, 29.4Kw in a BJ492Q gasoline engine

Figure 5 shows the calculation results with the change of the coefficient of the residual exhaust gas F_r, which is the ratio of the mass of the residual exhaust gas to the fresh mixture. When the range of F_r is 0.03～0.09, the change of the indicator diagram and indicated work is slight, and the change of the temperature and the velocity of the turbulent flame is also slight in both burned and unburned zones. However, the change of the concentration of NO_x is great. The concentration of NO_x reaches its minimum at F_r =0.09. So, Exhaust Gas Recirculation (EGR) and reduction of ignition advance angle can be used to reduce the explosion pressure and the maximum of the temperature in combustion. Meanwhile, these measures can reduce NO_x emitted with only little influence on indicated work and thermal efficiency.

6. Analysis of Calculation Results from the Multi-zone Model

Table 1 Changes of the temperatures in small zones (in a production BJ492Q with operating condition 2150 r/min, 34 kW)

Crank angle (°)	Temperatures in the unburned zone (K)	Temperatures in the preparing zone (K)	Temperatures in the burned small zone I (K)	Temperatures in the burned small zone Ⅱ (K)	Temperatures in the burned small zone Ⅲ (K)	Temperatures in the burned small zone Ⅳ (K)
343	713.3	2454.2				
349	737.7	2385.2				
355	764.9	2399.6				
361	801.0	2431.1				
367	849.7	2461.3	2461.0			
368	859.1	2458.3	2482.3			
369	869.0	2459.8	2504.3			
370	879.1	2468.6	2526.3			
371	888.8	2483.4	2547.3			
372	898.5	2483.6	2567.9	2483.4		
373	905.1	2514.2	2582.3	2498.4		
374	910.1	2538.4	2593.6	2510.0		
375	914.6	2548.5	2603.8	2520.5		
376	919.3	2551.8	2614.3	2531.3		
377	924.1	2552.0	2624.3	2542.2	2551.8	
378	927.8	2572.3	2633.3	2511.0	2560.6	
379	930.6	2590.7	2640.0	2557.9	2567.4	
380	932.7	2605.3	2645.2	2563.3	2572.8	
381	934.2	2616.9	2649.4	2567.6	2557.1	
382	935.3	2628.1	2652.5	2570.4	2580.4	
383	936.2	2628.1	2655.5	2573.9	2583.4	2628.1
384	936.4	2645.6	2656.9	2575.4	2584.9	2629.5
385	936.0	2662.9	2657.1	2575.6	2585.0	2629.7
386	934.9	2606.6	2655.9	2574.4	2583.9	2628.6
387	933.2	2520.2	2653.4	2571.8	2581.3	2626.0
388	931.0	2426.7	2649.8	2568.1	2577.6	2622.4
389	928.5	2334.9	2645.5	2563.6	2573.2	2618.0
390	926.0	2248.4	2640.6	2558.6	2568.1	2613.1
391	923.3	2188.5	2634.9	2552.8	2562.3	2607.4
392	920.3	2154.7	2628.5	2546.1	2555.7	2600.8

Table 1 shows the calculated results of the temperature in the cylinder from the multi-zone model. From the results, it is known that there are differences of the temperatures between the small zones. The temperatures in the small zones formed in early combustion are greater than those formed in late combustion, but the light-off temperatures in the latter are greater than those in the former. The reasons are that the latter compress the former, and the cylinder walls cool the latter at the end of the flame propagation. Furthermore, because of dividing the burned zone by the equipartition of mass, the volumes of the small zones formed in early and late combustion are greater than those formed in middle combustion, and the areas of heat transfer of the first and last zones are larger. Consequently, the temperatures of these two zones fall quickly. Therefore, the multi-zone model can describe combustion in gasoline engines more realistically.

7 Summaries

（1）Both a two-zone and a multi-zone model have been investigated for combustion in a gasoline engine. The calculation results agree well with experimental measurements, showing that the simulation is reasonable.

（2）To calculate the rate of combustion, the turbulent flame speed was applied, thereby improving the precision of the computations.

（3）The modeling identifies the important parameters which greatly affect an engine's performance. For example, the amount of NO_x emitted is very sensitive to the coefficient of residual exhaust.

（4）In the multi-zone model, the burned zone is divided into some small zones with different temperatures , so that the combustion in the cylinder can be described more realistically.

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