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Fast discrete resolution of combustion process in diesel engine based on cylinder pressure parameter

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1. Introduction

With the increasing concern over world energy resources and worldwide environmental issues, regulations for fuel efficiency and emissions on current engines are becoming more and more stringent. Closed-loop control is an effective method to precisely control fuel injection. Current diesel engines are controlled based on the rotational speed, which is an open loop control to combustion process because there is not a feedback parameter from combustion condition. Combustion information is key to condition flag of engine control, there is different gaining method of combustion information, cylinder combustion is treated by severval method to meet different calculation aim. Thermal-dynamic method is early calculation for combustion process, for instant, Austen-Lyn triangle menthod, White house single area method, this calculation result is rough to description of cylinder combustion, overall parameters, such as temperature and pressure of in-cylinder. Partion area calculation is relative progresss in contrast to thermal-dynamic method, because chamber space is divides into several areas according to time change, Cummins company Lyn combustion model of gas-phase fuel spray and combustion model of fuel drop evaporation are typical calculation method and applies widely. Three-dimension calculation is an accuracy method for fine model of fuel injection and fuel–air mixture and so on, its model calculation result is more suitable to real cylinder condition. FIRE, STAR–CD and KIVA software are current popular calculation means, its calculation result, such as portion temperature, emission NOx and PM, is precise, but its disadvantage is long calculation time, these are difficult to measure in real time. How to select simple easy-detection parameter is important to engine control. Cylinder pressure directly reflects combustion process in contrast with temperature and releasing heat, also includes combustion information. Introducing cylinder pressure into electronic controlled system can provide feedback of combustion process [1]. To abstract pressure characteristic index from in-cylinder pressure curve has become a key point of electronic control engine. In order to get combustion information quickly and apply it into engine control, an effective path is to obtain combustion characteristic parameter from cylinder pressure curve directly. Powell presented control spark timing by location of peak pressure (LPP) in spark ignition (SI) engine in order to make air-fuel ratio (AFR) to object value [2]. Mateunas et al presented a method of pressure ratio (PR) to obtain character of cylinder pressure curve, in order to made cylinder combustion to be best by mediating ignite timing in minimum spark timing for the best torque (MBT) [3]. Leondhart provided another method of forming dynamic injection quantity and injection advance angle by cylinder pressure, so that monitoring injection parameter is performed [4]. With the development of electronic control and sensor technology, it is possible to optimize combustion process by pressure information.

2. Combustion process calculation analysis

For four-stroke engine, one working cycle is made up of intake, compression and combustion working and emission stages, combustion process is happened in fixed stage in continuous working cycle, so combustion parameter changes with time, it belonged to continuous process. In the combustion process, combustion parameter variation is in high speed. Obvious character of combustion process in diesel engine is inhomogeneous distribution of working medium in cylinder, in the fuel spray formed by injector, distribution condition of fuel, air and combustion production is not only varied from time change, but also from different space position. Combustion process is un-linearity and transient change in several microseconds.

2.1. Combustion thermal process

In order to analyze changed condition and character condition, thermal process is a basic character in the complex physics and chemical change, when cylinder is regarded as thermal system, boundary is made up of piston top, cylinder cover and cylinder inter surface, as showed in Fig. 1, a typical thermal approach is used to describe condition change with pressure, temperature, mass and gas, thermal simulation equations are as followed.

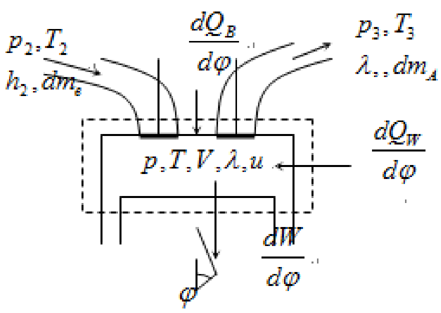

Fig. 1 Thermal-dynamic system of cylinder

Condition conservation equation

$$pV = mRT \tag{1}$$

Energy conservation equation

$$dU = \sum_i dE_i \tag{2}$$

Mass conservation equation

$$dm = \sum_j dm_j \tag{3}$$

where E_i is energy exchanged with system boundary, m_j is mass exchanged with system boundary [5].

This three equations are formed to resolve pressure, temperature and mass. In the four stroke diesel engine, mass conservation and energy conservation is followed with differential equation.

Intake process

$$\frac{dm_{in}}{d\varphi} = \frac{1}{\omega}\mu_E F_E \psi \sqrt{p_1 \rho_1} \tag{4}$$

Combustion process

$$\frac{dT}{d\varphi} = \frac{1}{mc_v}\left[\frac{dE}{d\varphi} - \frac{dU}{d\varphi}\right] \tag{5}$$

Releasing heat

$$\frac{dX}{dY} = 6.908(m+1)Y^m e^{-6.908Y^{m+1}}$$

$$\frac{dQ_B}{d\varphi} = H_u m_B \frac{dx}{d\varphi}$$

Making power

$$W_i = \int_{cycl} pdV = \frac{p_{i+1} + p_i}{2}(V_{i+1} - V_i) \tag{6}$$

Exhaust process

$$\frac{dm_{out}}{d\varphi} = \frac{1}{\omega}\mu_A F_A \psi \sqrt{p_4 \rho_4} \tag{7}$$

where message of all kind of above variable is listed in material [6] in detail.

2.2. Traditional resolution method of comsution process

For combustion process equation, there are three methods to resolve the in-cylinder parameter. Zero-dimensional model is simple method, it is called single area model or thermal model. All of parameters is purposed to only change with time and is neglected change with position, so there is only one independent variable time *t*. In the zero dimensional model, invariable differential equation groups are used to calculate temperature and pressure in cylinder. Quasidimensional model divided combustion chamber space into several areas, in the each area, multi differential equation groups are used to calculate temperature and pressure in cylinder, in order to predict emission character, as part temperature and air–fuel ratio are different in different position of combustion chamber, considered from combustion, ignite mixture forming and flame spread, partition model is developed. Multidimensional model is most complex treatment to thermal system, time, position, space of combustion chamber is considered, such as gas flow motion, mass and motion energy and energy transmission and conversion, is included, fuel injection, atomization and evaporation and fuel gas mixture forming, ignition and combustion, and heat transmission, gas matter and boundary motion (piston and gas valve) is described as submodel, so all energy conservation equation and submodel of turbulent flow, chemical reaction and boundary character are combined together, with suitable boundary condition, numerical method is used to resolve. Detail information of gas flow speed, temperature and composition in combustion chamber are provided, this model is relative fine one.

Time *t* is replaced with crankshaft angle θ in the differential equation groups, small fixed step $\Delta\theta$ is set, by means of numerical calculation method, iterated-interpolation calculation type and calculation precision is given, resolution of differential equation is obtained, in sum, this calculation time is from several seconds to several hours.

3. Cylinder pressure parameters

3.1. Fast resolution for combustion by cylinder pressure

When combustion process resolution is used to engine control application, there are many requirements for combustion model to meet control time, firstly it is fast calculation speed of combustion process, secondly, it is high accuracy of performance index provided by calculation method, finally, calculation result can meet change trend in transition profile. Combustion calculation time is less than that of control time of diesel engine, because in the profile of low speed and light load, time of each work cycle is longer and it is about 120~180 ms, in the profile of high speed and heavy duty, time of each work cycle is shorter and it is about 30~50 ms, so it is selected work cycle time corresponded to rated speed as evaluated criterion, on the contrary, combustion calculation time varied with adopted method, so that suitable method is selected to assure calculation time less than minimal work cycle time. Calculation parameter precision will affected the performance evaluation for electronic control system, the average error of output parameter is about 5%, maximal error of these parameters are less than 10%.

Combustion process is key to determine diesel engine combustion efficiency, combustion pressure is most direct and reliable measurement parameter of physical parameter in combustion process, and it can reflect multiinformation, as shown in Table 1, ignition commencement angle, maximum burst pressure and its angle vary with injection parameter, cylinder pressure change with energy conversion is corresponded, cylinder pressure curve can provide different character parameter in engine combustion, all these can indicate that pressure character is corresponded with combustion character .

Table 1
Physical meaning covered by pressure parameter

Data type	Parameter	Physical meaning
Cylinder pressure	Maximum burst pressure	Combustion intensity level
	Maximum burst pressure angle	Combustion maximum position
	Maximum pressure raising rate	Combustion increase level
	Maximum pressure raising rate angle	Combustion increase level position
	Average indicating pressure	Combustion overall level
	Ignition commencement angle	Combustion start to TDC
	Combustion last angle	Combustion last time
Heat releasing quantity	50%mass burn rate	Fuel quantity joining to combustion
	95%mass burn rate	Fully ignition fuel quantity
	90 heat releasing quantity	Fully heat releasing quantity

3.2. Combustion pressure index

Based on real measurement cylinder pressure, releasing heat index can be obtained from releasing heat rate or mass fraction burned by thermal dynamic calculation from cylinder pressure [7]. Calculation accuracy of the former method is determined by transient specific heat volume γ whose average value is usually given in the compression and expand stroke respectively, so calculation result is varied from different given value. This method requires high calculation configure, long calculation time and high demand for cylinder pressure sensor. The method based on mass fraction burned directly calculates cylinder pressure curve, which can decrease personal factor.

Pressure difference method invented by Bosch Company was used to analyze combustion condition. It is obtained by subtracting motor pressure from combustion pressure in the same work cycle, as shown in Fig. 2 is pressure difference curve represents results from fuel combustion injected into cylinder. Integral to pressure difference from commence angle to some angle after top dead center (TDC) represents combustion quantity. In the pressure difference curve, positions of combustion commencement point, maximal burst pressure and its angle are obviously identified. The method can decrease calculation quantity effectively and structural effects of different engines.

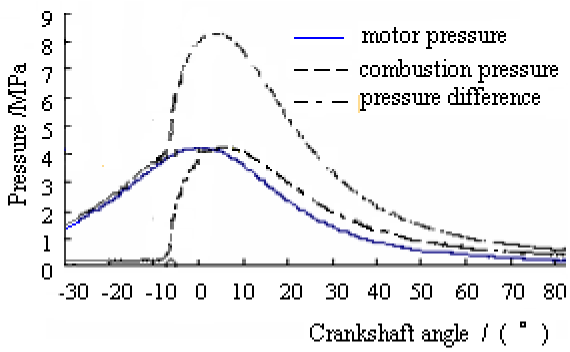

Fig. 2 Cylinder pressure curve

Mass fraction burned (MFB) is easy to calculate by pressure difference. Combustion process is indicated by dynamic thermal formula. Energy equilibrium of working medium in cylinder followed the first law of dynamic thermal mechanism

$$dW + dU = dQ \tag{8}$$

where dW is work done by piston ($dW = PdV$), dU is internal energy of work mass, and dQ is addition heat.

Addition heat quantity in combustion chamber is

$$dQ = H_u d(m_b) - d(Q_{wi}) \tag{9}$$

where H_u is lower heat quantity of fuel, m_b is fuel mass burned, and Q_{wi} is heat exchange with chamber walls.

Relation equations of an ideal gas are

$$dU = mc_v dT\ ,\ dT = \frac{V}{mR}dP\ ,\ \gamma = \frac{c_p}{c_v}\ ,\ R = c_p - c_v$$

where C_v is specific heat of given volume, C_p is specific heat of given pressure, and R is constant of ideal air.

dW, dU and dQ in Eq. (1) are replaced by cylinder pressure P and volume V and addition heat to yield following equation.

$$dP + \frac{\gamma\ P}{V}dV = \frac{R}{c_v}V\left[H_u d(m_b) - d(Q_{wi})\right] \tag{10}$$

When the differentials are expressed as derivatives with respect to crankshaft rotational angle, this equation represents a first order liner differential equation with following solution, cylinder pressure $p(\theta)$ is expressed by motor pressure $P_{mot}(\theta)$ in a given crankshaft angle θ.

$$p(\theta) = P_{mot}(\theta)[1.0 + (c_v m)^{-1}]INT1 \tag{11}$$

In the preceding equation, $INT1$ is the integral from ignition commencement angle θ_{fi} to combustion end angle θ_{fe} of the following expression

$$\left[H_u \frac{dm_b(\theta)}{d\theta} - \frac{dQ_{wi}(\theta)}{d\theta}\right]\left[\frac{V(\theta)}{V_0}\right]^{\gamma-1} d(\theta) \tag{12}$$

where $V(\theta)$ is cylinder volume of crankshaft angle θ, V_0 is combustion chamber volume.

By defining a term of the total heat of combustion, H_u, fuel injection m_f, and factoring it from within the integral, Eq. (1) may be rewritten

$$p(\theta) = P_{mot}(\theta)\left[1.0 + \frac{H_u m_f}{c_v m}\right]INT2 \tag{13}$$

In the preceding equation, $INT2$ is the integral from ignition commencement angle θ_{fi} to combustion end angle θ_{fe} of the following expression

$$\left[\frac{dm_b'(\theta)}{d\theta} - \frac{dQ_{wi}'(\theta)}{d\theta}\right]\left[\frac{V(\theta)}{V_0}\right]^{\gamma-1} d(\theta) \tag{14}$$

In integral *INT*2, the following terms apply, mass rate burned $m_b^{'}(\theta)$ and heat exchange rate $Q_{wi}^{'}(\theta)$ with chamber walls is followed

$$m_b^{'}(\theta) = \frac{m_b(\theta)}{m_f} \tag{15}$$

$$Q_{wi}^{'}(\theta) = \frac{Q_{wi}(\theta)}{H_u\, m_f} \tag{16}$$

The terms in Eq. (15) represents the mass fraction of fuel burned at any crankshaft angle, while the term in Eq. (16) represent the wall heat transfer as a fraction of total fuel energy.

Recognizing that the mass m of the combustion chamber content is the sum of intake air m_a, intake fuel m_f, and residual gasses m_r, the following equation applies

$$m = m_a + m_f + m_r \tag{17}$$

This allows Eq. (11) to be rewritten

$$p(\theta) = P_{mot}(\theta)[1.0 + A1A2(\theta)] \tag{18}$$

$$A1 = \frac{H_u}{c_v}\left[\frac{m_f}{m_a + m_f + m_r}\right] \tag{19}$$

$$A2 = INT2 \tag{16}$$

The factor $A1$ depends primarily on the mass of fuel relative to the mass of diluents, decreasing with greater dilution. The term $A2$ depends on the time of for combustion and the timing of combustion in the cycle along with the wall-loss rate.

Derivation process is not listed in the paper due to the limit of page number. Final derivation result is written as

$$M_{fb}(\theta) \approx \frac{P_{fire}(\theta) - P_{motor}(\theta)}{P_{motor}(\theta)} = \frac{\Delta P_{cd}(\theta)}{P_{motor}(\theta)} \tag{20}$$

where $M_{fb}(\theta)$ is mass fraction burned, $P_{fire}(\theta)$ is combustion pressure of fuel-air mixture, $P_m(\theta)$ is compression pressure of fresh air, and ΔP_{cd} is pressure difference of a given crankshaft angle .

Conclusion is drawn from Eq. (20). In each crankshaft angle, MFB is a ratio of pressure difference and motor pressure. So there is a relationship between releasing heat and MFB.

3.3. Combustion characteristic parameter

Combustion pressure data are mainly used for design purpose and combustion analysis. Releasing heat indexes obtained by thermal dynamic calculation based on cylinder pressure are used to evaluate combustion reasonableness [8]. As combustion process repeats in periodic work cycles, we introduced phase and intensity of physical parameter in periodical motion to describe combustion process. Combustion character parameter covers combustion phase, combustion intensity and releasing heat prompting in cylinder. It includes combustion commencement and end position, maximum burst pressure, maximal pressure rising rate and its angle, 50% MFB, 95% MFB and maximum releasing heat quantity. The parameters reflecting combustion phase include maximum burst pressure angle, maximum pressure rising rate angle and 50% MFB angle. Maximal burst pressure, maximal pressure rise rate and 50% MFB reflect combustion intensity. Among these combustion parameters, character parameters used for combustion control must be comprehensively described character for combustion process. They must not only reflect combustion commencement position, intensity and releasing heat quantity, but also be accurate and need less acquirement time.

Pressure difference curve is an approximate smooth Gauss curve. Its geometry character covers start point, maximal top point, shape factor, bar-center point, first order derivative, and second order derivative. Among these character points, calculation quantity is different. To reduce calculation time, the geometry characteristic indexes of pressure difference curve are used to describe the combustion process (Fig. 3) such as maximal pressure difference ΔP_{max} and its angle $\Delta\theta_{Pmax}$, bar-center point (ΔP_C, $\Delta\theta_C$), combustion commencement and angle difference of fixed pressure.

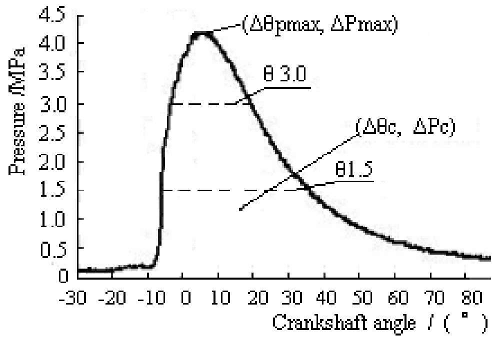

Fig. 3 Pressure difference characteristic index

4. Algorithm validity

4.1. Calculation of cylinder pressure data

Above analysis is made from the viewpoint of math and physics, since in-cylinder combustion is not only physical and chemical process, but thermal dynamic process, and is in nonlinearity complex change, it is needed to find the relation of engine input and character index. Cylinder pressure under the different condition of injection time and injection quantity is analyzed, due to limited experimental data, it can not meet requirement of acquisition analysis data in any profile, by means of GT-power simulation software, YN4100 diesel engine is selected as calculation object, its specification is followed: bore/stroke 100/105 mm, cylinder number 4, compression ratio 18.5, swirl ratio 2.5, rated power 70 kW, maximal torque/rotation speed 185 Nm/2200 r/min. Its combustion simulation model of is structured, one dimension simulation calculation is made, simulation model is shown in Fig. 4.

Model calculation accuracy is verified by experimental data, cylinder pressure, engine torque, engine power and fuel consumption is selected to evaluated index, in order to verify simulation model precise. Cylinder pressure of calculation and real measurement in the profile of engine rotational speed 2200 r/min is compared in Fig. 5. Seen from Fig. 4, the two curve trends are correspond in the most range though there is a little difference in small position, it is shown that engine combustion process and selected parameter is reasonable.

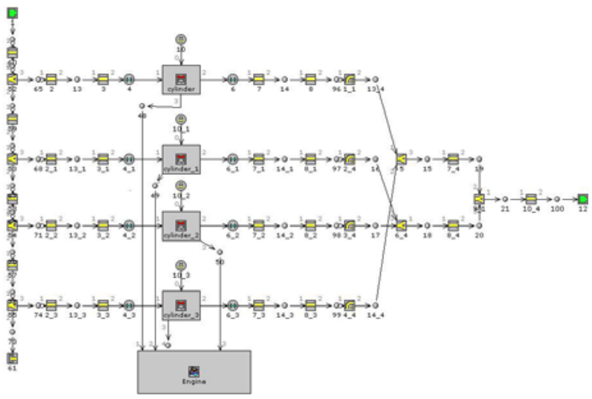

Fig. 4 Simulation model of YN4100 diesel engine

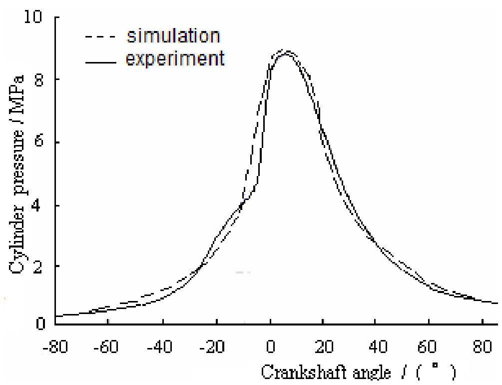

Fig. 5 Comparison of calculation and measurement pressure

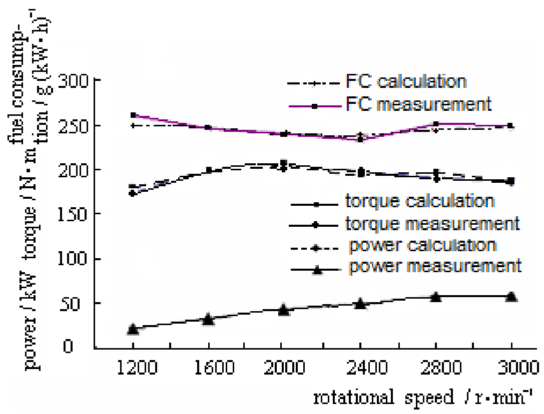

Fig. 6 Output performance parameter of diesel engine

It is chosen some profiles in the model calculation, which is rotation speed 1200, 1400, 1800, 2200, 2600, 3000 r/min, engine load 50%, 80%, 100% respectively, performance parameter of diesel engine is calculated by simulation model, output torque, output power and fuel consumption of calculation and real measurement are compared in Fig. 6. Comparison results in Fig. 6 indicate that calculation error for each parameter of calculation and real measurement is in less 5%. All this is shown that simulation model can reflect real combustion condition and model calculation precision is reasonable.

With above simulation model, some calculation point is chosen when engine rotation speed is set as 2200 r/min, its given parameter of simulation calculation is shown in Table 2, pressure character index of pressure difference is calculated in the different injection timing and injection quantity.

Table 2

Simulation calculation point parameter

Type point	Injection timing,°CA	Injection quantity, mg
0	0	0
1	10,13,15,18	10,20, 30, 40

4.2. Combustion characteristic parameter

To verify the effectiveness of the method, we use a DC motor as the controlled object to simulate in As engine combustion begins with drastic increase of cylinder pressure in cylinder pressure curve, pressure rise point before TDC is regard as fuel ignite position in the pressure difference curve, based on cylinder pressure calculated by simulation model, combustion commence angle calculated from pressure difference curve is shown in Table 3. Ignition commence angle is different from injection timing, and its position is related with injection quantity, this varying regularity is fit combustion rule.

Table 3

Ignition commencement angle change, °CA

Injection Timing, ° CA	Injection quantity, mg			
	10	20	30	40
10	7.5	8.1	9.0	9.8
13	9.0	9.8	10.5	10.9
15	11.2	12.4	12.8	13.6
18	15.2	16.1	16.8	17.1

Change of pressure character index under different injection timing and injection quantity is shown in Figs. 7 and 8. From Fig. 7, maximal pressure difference ΔP_{max} and its angle $\Delta\theta_{Pmax}$ was mapping one by one with each injection time and injection quantity, both of them did not exist intercrossing and overlapping. Maximal pressure difference ΔP_{max} was direct proportion to injection quantity and its angle $\Delta\theta_{Pmax}$ was constant in the same injection timing.

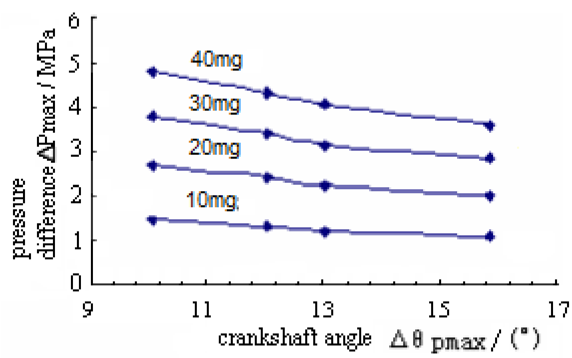

Fig. 7 Trend of maximal pressure difference index in different injection parameters

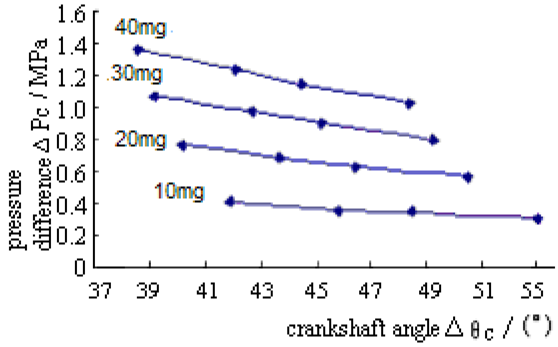

Fig. 8 Trend of bar-center point index in different injection parameters

From Fig. 8, maximal pressure difference ΔP_C varied directly with injection quantity. The angle $\Delta\theta_C$ was moved to forward with injection timing decrease from 18~10°CA. Each bar-center point consisted by ΔP_C and $\Delta\theta_C$ is corresponded to a given injection quantity and injection timing. There was no intercrossing and overlapping too. The results indicated that relation pressure character index has direct relationship with injection timing and engine profile parameter.

Output torque of diesel engine mainly varies with injection quantity and changes less with injection timing. In the character index of pressure difference curve, maximal pressure difference ΔP_{max}, pressure difference ΔP_C of bar-center point can reflect the change of injection quantity. Pressure difference ΔP_C of bar-center point can indicated overall level of combustion pressure change. The change of pressure difference ΔP_C of bar-center point with engine torque is shown in Fig. 9. From Fig. 9, in the condition of different injection quantity, engine torque increased linearly with ΔP_C increase. In the other condition of fixed injection quantity, engine torque changed less with injection timing increase. The change trend of maximal pressure difference ΔP_{max} with engine torque was basically similar with that of pressure difference ΔP_C.

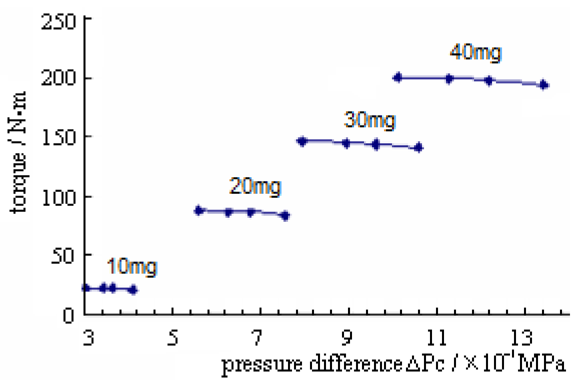

Fig. 9 Change of pressure difference ΔPc and torque

Nitric oxide (NOx) and particulate matter (PM) are main emissions of diesel engine. NOx was initially selected as representative emission of diesel engine on preliminary stage. Changes of rotational speed and fuel injection timing could directly affect NOx production. ΔP_{max} could reflect variety of injection timing and position relation between maximal burst pressure and TDC. Due to coming time of maximal temperature lagging to that of maximal burst pressure, ΔP_{max} could directly represent coming position of maximal temperature. Experimental data were used because of some errors of NOx and PM calculation combustion model. Rotational speed 1200, 1400, 1800, 2200, 2600, 3000 r/min were chosen and represented by 1, 2, 3, 4, 5, 6 respectively in Fig. 10. NOx concentration of 4100 diesel engine was measured in the bench, the relation of NOx concentration and maximal pressure difference ΔP_{max} is shown in Fig. 10. From Fig. 10, at same rotational speed, when engine load increased, NOx concentration basically increased with maximum value of pressure difference. When injection quantity was enlarged, maximal burst pressure increased with injection quantity. In the range of engine load 50%, 80%, increasing argument of maximal pressure difference is much more so that NOx concentration produced more, but in high engine load 80~100%, increment of maximum pressure difference was not more, and NOx concentration produced less.

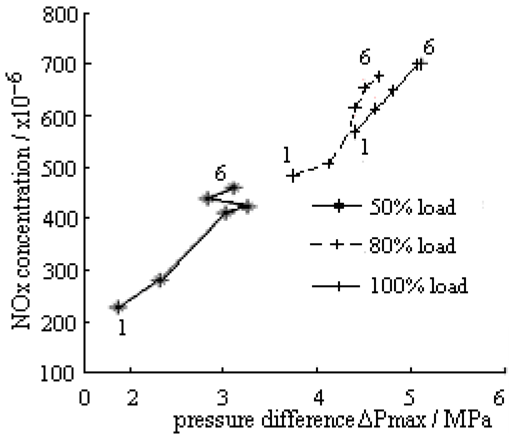

Fig. 10 Change of pressure ΔP_{max} and NOx

5. Conclusion

Fast discrete resolution to combustion process is presented to simplify calculation and save calculation time, combustion intensity, combustion phase and releasing heat were provided to describe combustion condition. Pressure difference method was introduced into combustion process to analyze cylinder pressure. Geometry character values were abstracted from pressure difference curve: ignition commence angle, maximal burst pressure and its angle, bar-center point and angle difference of fixed pressure, several pressure character parameters were used to specialize combustion overall character. Relation of pressure character index with injection timing and quantity was studied by means of simulation and measurement. Each pressure character index corresponded with a given emission value of diesel engine. The results indicated that fast discrete resolution for combustion process with pressure parameter was effective and pressure character index could reflect combustion character.

Acknowledgment

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GREITAS DISKRETUS ĮPURŠKIMO PROCESO DYZELINIAME VARIKLYJE SPRENDIMAS PAGAL CILINDRO SLĖGIO PARAMETRUS

R e z i u m ė

Įpurškimo informacija reikalauja, naudojant variklio kontrolės sistemą, įprastus įpurškimo proceso skaičiavimo metodus analizuojant glaustai, įpurškimo sąlygas apibūdinti cilindro slėgio skirtumo metodu. Cilindro slėgis naudojamas kaip įprastas rodiklis norint gauti įpurškimo informaciją remiantis cilindro slėgio parametrais, taikant slėgio skirtumo metodą ir žinomą greitą įpurškimo proceso slėgių skirtumo sprendimo matematinį algoritmą. Įpurškimo intensyvumas, fazė ir karščio mažėjimo greitis naudojami įpurškimo sąlygoms nusakyti. Uždegimo momento kampas, maksimalus sprogimo slėgis ir jo kampas bei slėgio centro taškas slėgio kitimo kreivėje yra išrinkti kaip būdingi parametrai, nusakantys bendrą įpurškimo pobūdį. Kartu su realaus variklio slėgiu slėgio pobūdžio indekso ryšys su įpurškimo laiku ir kokybe buvo lemiami imituojant ir matuojant slėgio duomenis ir keičiant kiekvienos slėgio charakteristikos pobūdį, remiantis eksperimento duomenimis, priklausomai nuo dyzelinio variklio NO emisijos. Rezultatai rodo, kad greitas ir protingas įpurškimo proceso sprendimas, naudojant slėgio parametrus, buvo efektyvus ir slėgio pobūdžio indeksas gali apibūdinti įpurškimo charakteristikas.

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FAST DISCRETE RESOLUTION OF COMBUSTION PROCESS IN DIESEL ENGINE BASED ON CYLINDER PRESSURE PARAMETER

S u m m a r y

Combustion information is needed to apply engine control system, current calculation methods for combustion process are analyzed briefly, cylinder pressure difference method was introduced to study combustion condition. Cylinder pressure is used to be combustion indication flag in order to meet combustion information requirement, based on cylinder pressure parameter, by means of pressure difference method, math algorithm of fast dispersed resolution to combustion process by pressure difference is presented. Combustion intensity, phase and heat-releasing rate are provided to describe combustion condition. Ignition commencement angle, maximal burst pressure and its angle, and bar-center point in pressure difference curve are selected as character parameters to specialize overall combustion character. Combined with cylinder pressure of real engine, relation of pressure character index with injection timing and injection quantity was resolved by simulation and measurement of pressure data, and change trend of each pressure character index with diesel engine emission NOx based on experimental data was analyzed. The results indicated that fast discrete resolution for combustion process with pressure parameter was effective and pressure character index could reflect combustion character.

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