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ENGINE CONTROL USING COMBUSTION MODEL

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ABSTRACT−The combination of physical models of an advanced engine control system was proposed to obtain sophisticated combustion control in ultra-lean combustion, including homogeneous compression-ignition and activated radical combustion with a light load and in stoichiometric mixture combustion with a full load. Physical models of intake, combustion and engine thermodynamics were incorporated, in which the effects of residual gas from prior cycles on intake air mass and combustion were taken into consideration. The combined control of compression ignition at a light load and spark ignition at full load for a high compession ratio engine was investigated using simulations. The control strategies of the variable valve timing and the intake pressure were clarified to keep auto-ignition at a light load and prevent knock at a full load.

KEY WORDS : Engine, Combustion, Electronic control

1. INTRODUCTION

There are many strategies for improving fuel economy and reducing exhaust emission. Hydrocarbon emissions (HC) and carbon monoxide (CO) are reduced using catalyst. Reducing nitrogen oxides emissions (NO_x), ultra-lean combustion, including homogeneous compression-ignition and activated radical combustion, seems promising for the near future (Ohyama, 1999). In these systems, combustion takes place in a narrow region between extinction and ignition. Therefore, the implementation of more sophisticated physical models than those in conventional control systems is necessary. But, model on engine thermodynamics and combustion in the condition of high air fuel ratio (A/F), high exhaust gas re-circulation (EGR) and variable compression ratio (CR) have not yet been completely clarified (Ohyama *et al*., 1998, 1999). In this paper, the combination of an intake model, combustion model and engine thermodynamic model was investigated to accurately estimate intake air mass, burn rate and auto-ignition delay. The combination can be applied to combustion, not only with flame propagation such as conventional spark ignition, but also with auto-ignition such as homogeneous compression-ignition.

2. ENGINE CONTROL STRATEGIES

There are many control strategies in engine control to improve fuel economy and reduce exhaust emissions, including the following.

– Gasoline engine: variable valve timing, improved ignition systems, adapted boost, direct injection stratified charge, controlled combustion, cylinder deactivation, divided chamber stratified charge are included.

– Diesel engine: direct injection, exhaust turbocharger are included.

– Ultra-lean combustion engine (homogeneous combustion or activated radical combustion):

In conventional direct injection diesel engines, direct injection gasoline engine combustion takes place at a stoichiometric contour, generating the highest temperature the fuel can attain with air, irrespective of the air-fuel ratio in the cylinder. The execution of its exothermic process with locally stoichiometric composition of air-fuel mixture is a natural kind of the process, such as fire, resulting in high NO_x. Conventional compression ignition engine suffers from high soot fromation.

2.1. Ultra-lean Combustion

For reducing NO_x, ultra-lean combustion is promising (Ohyama, 1999; Pascal, 1999). Ultra-lean combustion in industrial combustion furnaces using preheated air of 1273 K (Tokumoto, 1994) and a porous body (Echigo, 1996) has been presented. Generally, ultra-lean combustion may be attained by:

(a) catalytic combustion
(b) radical addition
(c) preheat (preheated air of 1273 K, fuel direct

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injection)

(d) high pressure, compression ignition

(e) self ignition by elite nucleus

(f) ignition by supplementary injection of high cetane fuel

(g) jet ignition (Lavinia Activatsia Gorenia, lean homogeneous mixtures in bulk gas supplied by direct fuel injection, with distributed ignition from highly reactive jets)

(h) counter-acting jets of air-fuel mixture

(i) atmospheric thermo-active combustion (ATAC)

(j) Toyota-Soken combustion process

(k) pulsed jet combustion (PJC)

(l) self-ignition triggered by radical injection (APIR) (Pascal *et al.*, 1999)

ATAC requires high levels of combustion product recycling. ATAC design is more adapted to the two-stroke engine. The key to the homogenous premixed charge compression-ignition engine and activated radical combustion is retarding the appearance of the hot flame until the appropriate time by controlling the lifetime of low temperature products, such as HCHO (aldehyde). It is necessary to get conditions for retardation. The use of the exhaust gas recycle (EGR) slows down the speed of chemical reactions, which retards the start of ignition, and reduces the rate of combustion. Rapid and complex combustion control may be adopted by using the combustion models (Ohyama, 1999). The implementation of more sophisticated physical models than those in the conventional systems seems to be necessary. For example, feed back control by using an in-cylinder pressure sensor is promising. The control using the information about the combustion in prior cycle causes one cycle delay. To avoid the delay, the model based control is effective.

2.2. Advanced Control Stratrgies

The engine control strategies presented in this paper is as follows. With a light load, homogeneous charge compression ignition (Thring, 1989) or efficient light load auto-ignition combustion (Duret, 1996) is executed. To control auto-ignition, it is necessary to retard thermal reaction until an appropriate period by controlling the lifetime of low temperature oxidation products. The use of EGR slows down the speed of chemical reactions, which retard ignition, and reduces the rate of combustion (Christensen, Johansson, 1998). Also, auto-ignition can be controlled by CR. EGR and CR are controlled by variable intake valve timing. CO and HC are reduced using oxidation catalysts.

With a full load, compression ignition can not be controlled without heavy EGR or low CR, which reduce air mass in the cylinder, resulting in reducing engine power. To increase power, a homogeneous mixture is ignited with a spark plug or jet flame under a low compression ratio. By using cold EGR instead of fuel enrichment, a stoichiometric charge can be used at full load, and hence emissions of CO and HC are greatly improved (Grandin *et al.*, 1998). Furthermore fuel efficiency is greatly improved. To increase the air mass for the constant cylinder volume, the EGR mass is reduced and supercharging is increased using variable turbine geometry. CR is reduced to prevent knocking. The stoichiometric charge is ignited by a spark plug or jet flame as mentioned above. Auto-ignition can also be controlled by residual gas (Willand *et al.*, 1998).

3. COMBINATION OF PHYSICAL MODELS

3.1. Physical Model

One new approach which could help considerably in reducing the dependence on control maps in engine systems is to use physical models as the basis for control. To control the combustion without delay, it is necessary to estimate the combustion process of the next cycle. using the modles. The models of engine thermodynamics and combustion have not yet been clarified completely (Ohyama, 1999). In this paper, the combination of the intake manifold model and the combustion model was investigated. The models are combined as shown in Figure.

3.2. Combination of Models

Figure 1 shows a block diagram relating engine variables and parameters on cycle by cycle variations. Global nonlinear dynamics arising from the effects of residual gas from prior cycles on combustion are taken into consideration by combining the intake model, combustion model and engine thermodynamic model. This combination seems to result in an accurate estimation of auto-ignition delay and burn rate. More inferences can be

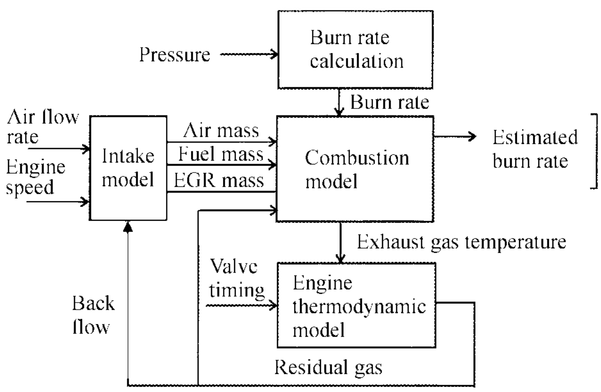

Figure 1. Block diagram combining intake model, combustion model and engine thermodynamic model.

made about the effect of engine variables on cycle by cycle variations.

The exhaust gas temperature is changed by the burn rate. The residual gas mass is changed according to the exhaust gas temperature and valve timing. The residual gas in the cylinder at the end of the exhaust stroke influences the dynamics of the mass flow rate into the cylinder in the intake system. The internal EGR modifies the cylinder intake capacity because the pressure in residual exhaust gas is different from the pressure in the intake manifold, especially at part load. The residual air mass and residual gas mass affect the burn rate of the next cycle. Therefore, it is necessary for the estimation of the burn rate and auto-ignition to combine the three models mentioned above.

3.3. Engine Thermodynamic Model

The residual gas in the cylinder at the end of the exhaust stroke (phenomena known as internal exhaust gas re-circulation- internal EGR) influences the dynamics of the mass flow rate into the cylinder. The internal EGR modifies the cylinder intake capacity because the pressure in residual exhaust gas is different from the pressure in the intake manifold, especially at part load. In automotive practice, this behavior is taken into account with the experimental determination of engine volumetric efficiency. An estimation of the internal EGR by a physical model (Ohyama, 2000) is presented so as to realize in the complete intake model the evaluation of the air mass flow rate required by the engine without the necessity of knowing the volumetric efficiency (Azzoni *et al.*, 1998; Nieuwstandt *et al.*, 1998). The connection between the thermodynamic conditions in the intake manifold, and those in the cylinder depend on the fluid flow motion through the intake valve and on the heat exchanges through the cylinder wall and between the air and the residual exhaust gas (Suzuki *et al.*, 1999). Techniques like exhaust gas re-circulation and turbo-charging have been devised to face the string of requirements on performance, fuel economy and emissions (Nieuwstandt *et al.*, 1998). As a result, the accurate estimation of the air mass and residual gas mass is necessary to improve engine performance.

The homogeneous exhaust gas of lean combustion contains excess air. The retained excess air further dilutes the lean charge. Homogeneous combustion with high ratios of retained exhaust gas burns leaner internally than is measured externally. The effect increases with higher amounts retained and leaner mixtures. The A/F in the charge and measured in the exhaust gas is transformed to the internal A*/F, which is a function of A/F and retained exhaust mass fraction. For combustion, A*/F and the retained inert mass fraction are controlled (Willand *et al.*, 1998).

In this paper, the parameters (residual gas, residual gas fraction, residual air mass, residual gas mass, residual fuel mass, the air mass in the cylinder at the end of intake stroke and the back flow at the beginning of the intake stroke) are calculated using the equations in Appendix B of Ref. (Ohyama, 2000). The A/F of exhaust gas, the residual fuel component, the exhaust fuel component, the residual air component and the exhaust air component are calculated by using the equations in Appendix C of Ref. (Ohyama, 2000). The A/F is more than the stoichiometric value in the above equations. These equations are integrated in the intake model, which has been described in Appendix A of Ref. (Ohyama, 2000).

3.4. Combustion Model

Engine operation increasingly employs sophisticated control with closed-loop operation of various functions. The NO_x control scheme based on cylinder event feedback has been presented (Challen and Stobart, 1998). Ignition timing control, knock control and each cylinder air mass control are executed using combustion pressure. The examination of the combustion pressure, as well as the thermodynamic calculation of the energy conversion computed from it, are used for the evaluation because of their high degree of accuracy. Using this model, the combustion process is divided into three periods: the inflammation period, the main conversion period and the post burning period.

The computer for on-board management, which is used to control an engine, is not capable of handling thermodynamic examination. Reaction from one cycle to the next is impossible because of the extensive calculations it requires. Therefore, a calculation model has been developed which allows the ready calculation of a value whose usability and accuracy are similar to the point at which 50% of the available energy has been converted, designated in degrees of crank angle (Gebauer and Muller, 1994). Data processing such as return maps, symbol sequence statistics, modified form of Shannon entropy (Wagner *et al.*, 1998), moments technique (Arisie *et al*., 1998; Jackson *et al*., 1996) and pressure measurement at crank angles of 90, 60, 40 deg. (degrees) BTDC (before top dead center) (Held and Scubert, 1994) has been presented. For example, load variable (air mass) can be estimated using the pressure at a crank angle of 90 deg. BTDC. For example, NO_x is controlled by using the combustion model, where NO_x is measured by using a NO sensor (Kato *et al.*, 1996) and compared with the estimated value in the model. Then, the burn rate can be modified to limit NO_x.

In the combustion model, the characteristic values are combustion pattern, combustion duration and the start of combustion. Mean effective pressure is calculated without numerical integration by using only the amplitude and phase of the fundamental and second harmonics components from the in-cylinder pressure signal. The in-cylinder pressure signal is transformed to Fourier coefficients of the series during the engine cycle (Appendix D of Ref. (Ohyama, 2000)). Then, the Fourier coefficients of the series are converted to the effective pressure and characteristic values of the combustion (Ohyama, 1999). The start of combustion θ_i and the combustion duration θ_d can be determined easily without numerical integration.

In this paper, the combustion model described in Appendix E of Ref. (Ohyama, 2000) is used. The model is a simple two-zone model. The approach developed avoids the prediction of detailed features of turbulent flame propagation. Auto-ignition delay (Tomita *et al.*, 1994; Mori *et al.*, 1998) is calculated using the equations in Appendix F of Ref. (Ohyama, 2000). The auto-ignition occurs when the Livengood-Wu integral becomes 1. At this time, knocking occurs when combustion takes place in the manner of flame propagation. Combustion starts when combustion takes place in the manner of homogeneous charge compression ignition.

3.5. Calculation Procedure

Figure 2 shows the calculation procedure for one cylinder of a four- cylinder engine. At the end of the intake stroke, air mass W_e is determined by inputting the air flow meter signal into the intake model. The air mass G_a, fuel mass G_f , total mass G_t and the air fuel ratio in the cylinder *A*/F* are calculated using the engine thermodynamic model. The parameters are inputted into the combustion model. Then the exhaust gas temperature T_{ex} at the exhaust stroke is calculated. Also, the parameters auto-ignition such as the delay time and the integral function are calculated. The residual gas ratio RES is a function of intake and exhaust valve timing. The residual gas fraction REX is a function of RES, exhaust gas temperature and pressure. RES is given in the engine thermodynamic model and the residual air mass G_{ra}, residual inert gas mass G_{rg} and residual fuel mass G_{rf} are calculated. These parameters are inputted in the intake model of the next cycle.

To simplify the calculation, at the beginning of the intake stroke, part of the residual gas is flowed back from the clearance volume to the intake manifold. The air mass in the cylinder contained the residual gas flowed in the intake manifold at the beginning of the intake stroke. Therefore, at the end of intake stroke, the air mass is compensated as described in Appendix B of (Ohyama, 2000).

4. ANALYSIS

4.1. Simulation Conditions

Table 1 shows the simulation conditions. A 4-cylinder, 4-stroke engine with total cylinder volume of 1500 cm^3 and a conventional intake system was used for the simulations. Fuel properties were shown in Appendix F of Ref. (Ohyama, 2000).

4.2. Misfire

When misfire occurred at the lean limit of *A/F*, the residual gas contained part of the unburned fuel. As a result, the *A*/F* of the next cycle became lower when *A/F*

Table 1. Simulation conditions.

Stroke volume	375 cm^3
Clearance volume	37.5 cm^3
Number of cylinder	4
Stroke	4
Engine speed	1200 rpm
Heating value of fuel	4.2×10^4 J/kg
Volume of intake manifold	1×10^{-3} m^3

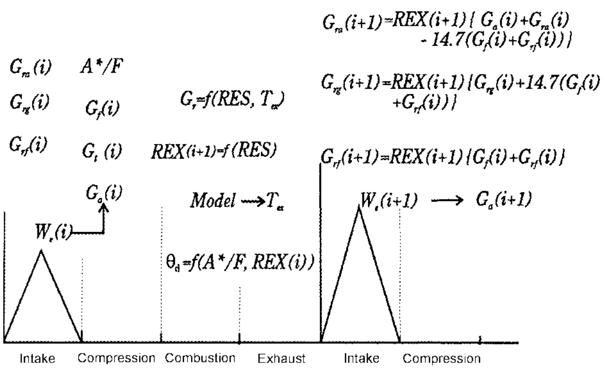

Figure 2. Calculation procedure.

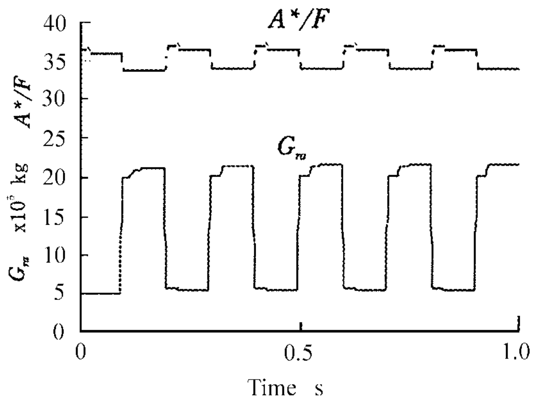

Figure 3. Traces of *A*/F* and G_{ra}.

was controlled constant and the combustion took place again due to rich mixture. After that, the residual fuel mass became zero again, the *A*/F* of the next cycle became higher and the misfire occurred again. Figure 2 shows an example of the cyclic misfire. When *A/F* was 31.7 and the throttle was wide open, θ_d was 60, *RES* was 0.5 and lean burn limit was 35, the *A*/F* became 36.15 without misfire. Prior to misfire, however, *A*/F* became 34.96. Then, misfire and combustion occurred cycle by cycle in the condition, as shown in Figure 3. Residual air mass G_{ra} increased when misfire occurred. The air mass of each cylinder varied slightly due to the change of mixture charge in the manifold as shown Figure 3. This cyclic misfire can be prevented by controlling G_f so that *A*/F* is not in excess of the lean burn limit The cyclic misfire can be easily observed in 2-stroke engines, in which the residual gas is higher than 4-stroke engine at partial load and fuel remains in residual gas when misfire take place.

The misfire in homogeneous compression ignition causes the increase in HC as the combustion stops due to insufficient heat at local lean mixture. To prevent the misfire, the mixture homogeneousness must be controlled precisely.

4.3. IN-Cylinder Pressure and Burn Rate

The characteristic values about the burn rate are the ignition (combustion) start θ_i and the combustion duration θ_d. The pressure signal is transferred to Fourier series. The effective pressure is a function of fundamental component and the 2nd harmonics of Fourier series. Figure 4(a) shows relationships between the Fourier coefficient of 0.5 higher harmonics $a_{0.5}$; the coefficient of the 2nd harmonics a_2; and the coefficient of fundamental wave $a_{1,}$ when θ_i is 170, 180, 190 deg. and the duration θ_d is 20, 40, 60 and 80 deg. As shown here, a_2 versus a_1 is independent of θ_d and θ_i. Therefore, the mean effective pressure is calculated from a_1 only, without integrating the pressure signal during the strokes. The characteristic values of combustion, θ_d and θ_I can be calculated by using the Fourier coefficients $a_{0.5}$ and a_1, using the relationships of the Figure 4(a). The burn rate can be determined easily without numerical integration, using the coefficients. The burn rate can be controlled to the target value using EGR which can be controlled by estimating the burn rate mentioned above. Also, the auto-ignition can be controlled using EGR, or variable valve timing control system.. The heat transfer through the cylinder wall has a great influence on the combustion process. But it is difficult to simulate the heat transfer precisely on the computer for on board management. The heat transfer is compensated using pressure signal.

Figure 4(b) shows the traces of burn rate calculated from pressure history and pressure calculated for a given

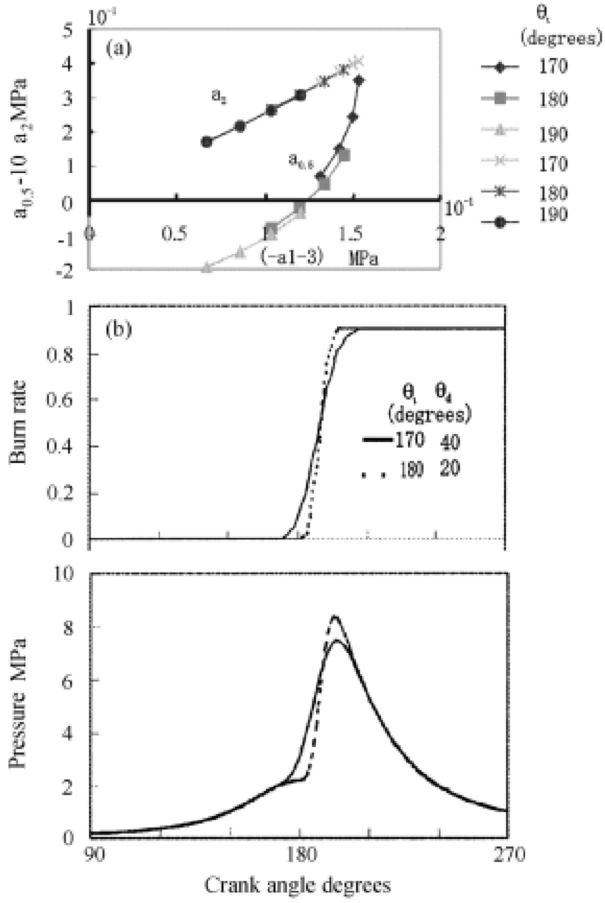

Figure 4. (a) Relationship between Fourier series, (b) Traces of burn rate and pressure.

burn rate as a function of crank angle θ. The burn rate is varied according to the ignition start θ_I and the combustion duration θ_d.

4.4. Effect of Cylinder Volume V_{cc} on Auto-ignition

Figure 5 shows the the Livengood-Wu integral I_{lw} and the fuel mass $\underline{G_f}$ versus the cylinder volume V_{cc} when the clearance volume was constant the compression ratio is 14. When the intake pressure was 100 kPa and *A*/F* was 15, I_{lw} becomes 1 at smaller V_{cc} than that of *A*/F* =20 and 22. When the *A*/F* increases from 15 to 20 and 22 at the pressure of 100 kPa, V_{cc} when I_{lw} is 1 increases, but G_f changes from q3 toq1 and q1, it does not change significantly when I_{lw}.is 1. The engine power remains same when the the integral I_{lw} is 1. As knock can be avoided to keep I_{lw} less than 1, by decreasing in the V_{cc}, when the *A*/F* decreases from 22 to 15. The cylinder volume V_{cc} can be controlled using the variable valve timing system. But, the engine power does not increase because G_f remains the same when *A*/F* decreases. The

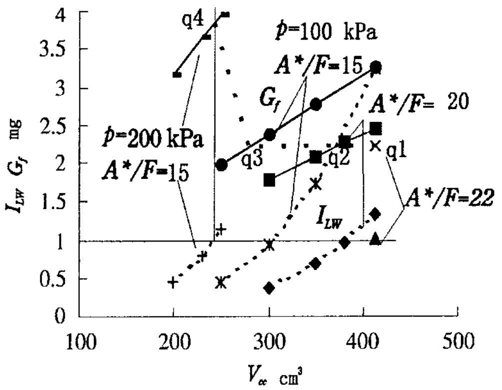

Figure 5. Traces of I_{lw} and G_f.

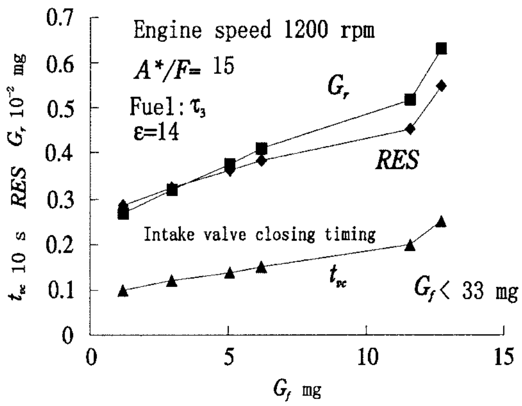

Figure 6. Control using variable valve timing.

increase in the fuel mass due to smaller A^*/F cancelled by the decrease in V_{cc}. When the A^*/F is 15 and the pressure is 200 kPa, I_{lw} becomes 1 at much smaller V_{cc}. But, the G_f increases significantly, as the intake pressure becomes higher. The engine power recovers to the level of a natural aspirated engine with low compression ratio. The high intake pressure engine with a turbo-charger or super-charger outputs more power without knock, when the intake air is cooled downstream of the charger. The knock occurs easily without cooling. The compression ignition at part load requires the high compression ratio, which is apt to induce knock at full load. It is necessary to reduce the compression ratio, but the mechanisms for variable compression ratio are complex. Therefore, the control of the effective compression ratio using the variable valve timing is used. But, the air mass decreases as the compression ratio decreases. It is necessary to boost using the charger. Knock varies according to the fuel properties. It is difficult to estimate the properties directly. Therefore, the combustion model is compensated using the pressure signal.

4.5. Control Using Variable Valve Timing

Figure 6 shows the residual gas mass G_r, the residual gas ratio *RES*, the intake valve closing timing t_{vc} versus the fuel mass G_f, in case of compression ignition where the integral I_{lw} is 1 at crank angle of 170 deg. (10 deg. before top dead center). The compression ignition takes place when the temperature of mixture is much higher than spark ignition. When the intake valve is closed early, resulting in the decrease in the fresh air mass and in the increase of the temperature of the mixture heated by the residual gas, the compression ignition take place. As the fuel mass increases, the air mass must be increased to keep the A^*/F constant. The valve timing t_{vc} is delayed as the G_f increases. Because the increase in the fresh air mass reduces the mixture temperature, it becomes difficult to keep auto-ignition as the G_f. increases. To increase the temperature under the high fresh air mass, the RES must be increased. The RES can be increased as the mass of residual gas increases using the early exhaust valve closing or early intake valve opening. In Figure 6, the maximum G_f in case of the spark ignition without super-charger is 33 mg. The maximum G_f in this compression ignition is 13. Therefore, The compression ignition by variable valve timing is possible only at part load.

The fuel mass G_f is determined by accelerator pedal position, taking the thermal and mechanical efficiency of the engine in consideration. Then, the intake valve timing is calculated using the physical models shown in Figure 1, which are compensated in real time by using the signals of the air flow meter and the in-cylinder pressure sensor. The RES is controlled by a valve installed in the exhaust system. The RES can be increased by closing the valve. When the auto-ignition does not take place at the target crank angle, the models are compensated, which can modify the models to the fuel properties and the heat transfer.

When the *A/F* changes, the temperature at the end of the exhaust stroke also changes. Therefore initial temperature of mixture during the compression stroke changes, which affects the auto-ignition start crank angle. The combination of combustion model and the intake model can estimate the temperature change when the *A/F* varies widely.

To control the compression ratio by an auxiliary piston installed on each cylinder is effective to control the temperature, auto-ignition and knock. It can control the temperature and the pressure in the cylinder independent of the intake system and the exhaust system. But the control mechanisms are complex.

The homogeneous compression ignition by the auto-ignition take place only at partial load. At full load, it is necessary to shift the combustion mode from homogeneous compression ignition to homogeneous spark ignition.

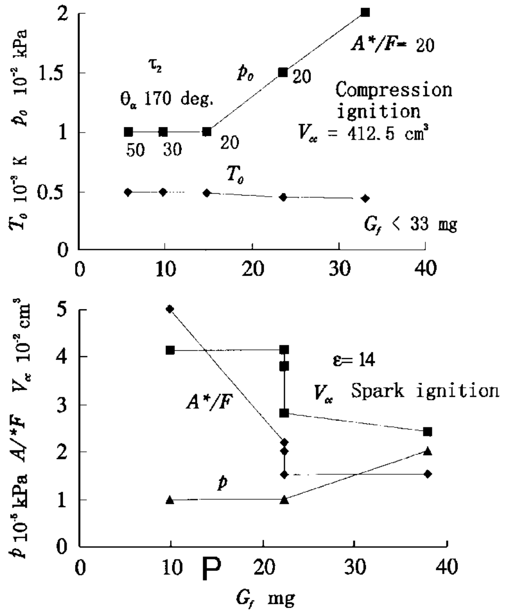

Figure 7. p, A^*/F and V_{cc} versus G_f.

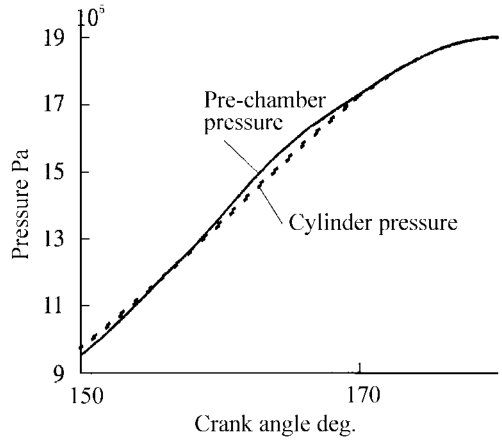

Figure 8. Traces of pre-chamber pressure.

The auto-ignition delay time of fuel properties τ_3 is 0.1 s at the temperature below 700 K and the pressure of 500 KPa.

4.6. Combination of Spark Ignition and Compression Ignition

Figure 7 shows the A^*/F and the effective cylinder volume V_{cc} and the intake pressure p versus the fuel mass G_f, without knocking, in case of spark ignition. The compression ratio ε was 14, the fuel type was τ_2. When the G_f increases, A^*/F decreases, V_{cc} must be decreased to reduce the effective compression ratio and prevent knock, resulting in the decrease in the engine power. To increase the air mass, fuel mass in the cylinder and the engine power, the intake pressure p is increased using a super-charger when the compression ratio is high enough.

Figure 7 also shows the intake pressure p and the intake temperature T versus the fuel mass G_f in case of compression ignition. V_{cc} was 412.5 cm^3, and the crank angle of ignition start was 170 deg. of BDC (bottom dead center). The compression ratio was 14. The temperature T remained high enough for auto-ignition to take place. Therefore, the air mass in the cylinder is smaller than that of the spark ignition. When the G_f increases and A^*/F decreases to 20, the air mass is increased to keep the A^*/F higher than 20, by using a super-charger. But, the maximum air mass and fuel mass are smaller than these of the spark ignition, because the initial mixture temperature is higher. Therefore, it is better to change the combustion mode from compression ignition to spark ignition to increase the engine power, at point P in Figure 7. To shift the point P at right side, the compression ignition driving region increased, resulting in lower NO_x, and lower fuel consumption, but the supercharger is driven at part load, resulting in the introduction of a complex control system. To shift the point P to the left side, the spark ignition region increased, resulting in the higher NO_xwhich must be purified using a catalyst. The shift point may be determined, taking the exhaust regulation and the requirement for fuel economy into consideration.

4.7. Pressure In Pre-chamber

In the self-ignition triggered by radical injection and the jet ignition the jet is injected from a pre-chamber to the cylinder. Figure 8 shows the traces of the pressure in the pre-chamber and the cylinder, calculated using the combustion model. The diameter and the length of the pre-chamber were 10 mm and 10 mm, respectively. The diameter of the pre-chamber orifice was 1 mm. The initial A/F in the pre-chamber was 10. There is no fuel in the cylinder at the start of compression stroke. The initial pressure in the cylinder and the pre-chamber was 98 KPa. The combustion of the mixture in the pre-chamber started at 150 deg. of crank angle. The duration of the combustion was 30 deg. The pressure in the pre-chamber is lower than the pressure in the cylinder before the start of the combustion during the compression stroke. The air in the cylinder is introduced into the pre-chamber during the compression stroke through the orifice. Some lean mixture flows back from the cylinder in case of jet ignition. The pressure in the pre-chamber becomes higher during the combustion. The incomplete burning of the rich mixture in the pre-chamber induces a rise of pressure and creates lots of intermediate combustion products. The combustion gas in the pre-chamber is ejected through the orifice into the cylinder. The pressure in the pre-chamber and the cylinder increases in the same manner after the combustion, during the compression stroke.

SUMMARY

Physical models of an advanced engine control system were investigated. By the combination of an intake model, a combustion model and engine thermodynamic model, the control of in-cylinder air/fuel ratio, misfire, knocking and auto-ignition was analyzed. The combination of the models can solve the nonlinear dynamics arising from the effects of residual gas from prior cycles on combustion and make more inferences about the effects of engine variables. The combined control of compression ignition at a light load and spark ignition at a full load for high compression ratio engine was investigated using simulations. The combustion model for engines with a pre-chamber was presented.

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NOMENCLATURE

A	: Air Mass
A_I	: Coefficient of Auto-Ignition Delay
a	: Cross-Sectional Area of Upstream Throttle
a_b	: Cross-Sectional Area of Bypass Valve
a_c	: Cross-Sectional Area of The Compressor
a_h	: Cross-Sectional Area of Throttle Valve
$a_{0.5}$, a_1, a_2	: Coefficient of Sine Component of Fourier Series
ah	: Coefficient of Cross-Sectional Area of Throttle Valve
B	: Coefficient of Auto-Ignition Delay
b	: Coefficient of Cosine Component of Fourier Series
c	: Specific Heat
F	: Fuel Mass
G	: Air Mass or Gas Mass
H	: Enthalpy
h	: Molar Enthalpy
l	: Length of Connecting Rod
l_c	: Effective Length of the Compressor
l_p	: Effective Length of Upstream Throttle
m	: Characteristic Value of the Wiebe Function
N	: Engine Speed
n	: Number of Moles
n_I	: Coefficient of Auto-Ignition Delay
ON	: Octane Number
p	: In-Cylinder Pressure
pai	: Pressure Ratio of the Compressor
$p_i(i)$	: Pressure in the Intercooler
p_m	: Intake Manifold Pressure
p_o	: Atmospheric Pressure
Q	: Heat
R_m	: Gas Constant of Mixture
R	: Gas Constant
RES	: Residual Gas Ratio
REX	: Residual Gas Fraction
r	: Radius of Crank
S_w	: Engine Displacement
T	: Temperature
T_i	: Temperature of the Intercooler
T_m	: Temperature of the Intake Manifold
t	: Time
u	: Internal Energy
V	: Volume
V_c	: Clearance Volume
V_i	: Volume of the Intercooler
V_m	: Volume of the Intake Manifold
W_a	: Air Flow Rate Through the Air Flow Meter
W_b	: Air Flow Rate of the Bypass Passage

W_e	: Air Flow Rate of the Compressor
W_e	: Air Flow Rate in the Cylinder
W_h	: Air Flow Rate Through the Throttle Valve
W_m	: Gas Mass in the Intake Manifold
W_r	: Exhaust Gas Recycle Rate
Z	: Burn Rate
η_v	: Volumetric Efficiency
κ	: Ratio of Specific Heats
θ	: Crank Angle
θ_d	: Crank Angle During Combustion Duration
θ_I	: Crank Angle at Start of Combustion
τ	: Auto-Ignition Delay
ξ_b	: Loss Coefficient of the Bypass Valve
ξ_c	: Loss Coefficient of the Compressor
ξ_h	: Loss Coefficient of the Throttle Valve

Subscripts:

a	: Air
b	: Burned Gas
bc	: Back Flow
ex	: Exhaust Gas
exa	: Exhaust Air Component
exf	: Residual Fuel Component
f	: Fuel
i	: Effective Mean
m	: Mixture
P	: Product
R	: Reactant
RP	: Combustion
r	: Residual Gas
ra	: Residual Air
rag	: Residual Air Component
rg	: Residual Inert Gas
rf	: Residual Fuel
rfg	: Residual Fuel Component
T	: Total Gas
t	: Total
u	: Unburned Gas
v	: Constant Volume
0	: Initial Condition
1	: First Step
2	: Second Step