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A Study of Squish in Open Combustion Chambers of a Diesel Engine*

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In the Diesel engine with an open combustion chamber, the utilization of gas motion produced in the chamber is important to improve engine performance. As the first step to finding the relation between the shape of combustion chamber and the formation process of squish, squish velocity has been measured with a velocity detector of solenoid type by motoring a model engine, and then investigated theoretically.

The measured squish velocity is not much different from the ideal one calculated under the condition without any loss. Through theoretical considerations, it is confirmed that the decrease of velocity due to both the heat loss to the wall and the leakage of charged air through the piston rings is remarkable in the period after 20 deg. before t.d.c. and becomes maximum at top dead center. However, the absolute value of decrement seems to be too small to affect seriously the squish velocity.

1. Introduction

The use of open combustion chambers for Diesel engines is gradually being extended to the range of higher speeds and smaller engine sizes, because of its high efficiency. In high speed engines, since the allowable time for mixture formation and combustion is short, one of the necessary conditions to improve the engine performance is the effective utilization of the gas motion produced in the chamber by squish or induced swirl. It has been understood that the toroidal motion, which is produced in the chamber by squish, is not so strong as to dominate the distribution of fuel spray. However, its auxiliary effect on mixture formation should not be ignored. The radial velocity at the brim of chamber occurring when the piston approaches the top dead center, that is, the squish velocity, is reduced by many factors such as pressure gradient due to gas inertia, friction, gas viscosity, heat loss to the wall, gas leakage through the piston rings, etc.^{(1) (2)} From the measured results of the intensity of gas motion in the chamber⁽³⁾, however, it may be estimated that the decrement of velocity is not so remarkable as to nullify the squish effect.

As the first step to finding the relation between the formation process and intensity of gas motion

and the shape of combustion chamber, the authors have measured the squish velocity by motoring a model engine in order to compare it with the ideal velocity calculated under the condition of no decrement. Furthermore, theoretical investigations have been carried out about the influences of both the heat loss and the gas leakage through the piston rings, which generate probably a relatively large decrement of the squish velocity.

2. Main notations

- A : heat equivalent to work
- c_m : mean piston speed
- c_v, c_p : specific heats at constant volume and pressure, respectively
- D : cylinder diameter
- d : diameter of combustion chamber
- F : area of wall surface exposed to the gas in the cylinder
- F_a : area of wall surface exposed to the gas which is contained between the annular flat-face of piston crown and the cylinder head
- G_a : weight of the gas contained in the space between the annular flat-face of piston crown and the cylinder head
- G_r : weight of the gas which leaks through the piston rings in compression stroke
- g : gravitational acceleration
- m : polytropic exponent
- n : engine speed, rpm
- p : pressure

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- Q : heat loss
 R : gas constant
 s : stroke
 T : absolute temperature
 T_w : absolute temperature of the wall
 t : time, sec
 u : squish velocity
 u_{th} : theoretical squish velocity
 V : volume in the cylinder
 V_a : volume of space between the annular flat-face of piston crown and the cylinder head
 V_h : stroke volume
 V_k : volume of combustion chamber
 Δs : top clearance
 Δu_h : decrement of squish velocity due to heat loss
 Δu_l : decrement of squish velocity due to gas leakage through the piston rings
 α : heat transfer rate
 γ : specific weight of gas
 θ : crank angle measured from top dead center, rad
 κ : ratio of specific heats
 λ : ratio of crank radius to length of connecting rod
 μf : effective flow area equivalent to the leaking area of piston rings

Subscripts

- 0 : state in reference (atmospheric pressure)
 c : state at beginning of compression

3. Experimental apparatus and method

The experiment has been carried out by motorizing a KAWASAKI KD101 four-stroke cycle Diesel engine (cylinder bore $\times$ stroke = 85 mm $\times$ 93 mm, normal output 10 PS/2 800 rpm). In order to make a model of open combustion chamber, the cylinder head is replaced with a flat plate, and a flat crown with a centrally placed cup which forms the combustion chamber, is bolted to the piston. Since the exhaust and inlet valves are removed, ports are provided at the lower part of cylinder to keep the cylinder pressure at bottom dead center at atmospheric level.

Fig. 1 Details of velocity detector

The compression and expansion of the gas are repeated by the piston in the cylinder without gas exchange. Only the amount of air equivalent to the leakage through the piston rings is supplied through the ports. The piston crown closes the ports at 100 deg. before t. d. c.

Hitherto, the gas velocity in the cylinder has been measured by utilizing hot wire anemometers⁽⁴⁾, or by detecting the deflection of thin plate, which is produced by the gas dynamic pressure, as change of inductance⁽⁵⁾ or capacitance⁽⁶⁾. The clever methods recently reported are to apply discharge of electricity for velocity measurement^{(7) (8)}. In this experiment the method to detect the velocity as an inductance change of solenoid is adopted from the point of view that the measuring equipment is simple and also the continuous recording of velocity is possible. Details of velocity detector are shown in Fig. 1. A fine needle, whose diameter is 0.4 mm, is soldered on the center of the pressure sensitive part of a thin steel plate. When the thin plate is deflected by the dynamic pressure of gas flow, the needle moves in the measuring solenoid. After being amplified, the inductance change of solenoid is displayed as a voltage change on the synchroscope as shown in Fig. 2. The filter is used to remove the carrier wave of 455 kHz. The magnetic permeability of gas changes with its density. Therefore, the solenoid which has the same size as the measuring one, is fitted to the detector to compensate for the inductance change occurring due to the compression of gas. The solenoids are inserted between the inner and outer ceramic tubes to keep their temperature distribution uniform, and further are shielded electrically by covering the outer ceramic tube with a thin steel tube. If the temperature distribution of solenoids is varied by the periodic change of gas temperature in the cylinder, the measured results are disturbed owing to the thermal effect.

Finding the gas density in the cylinder is necessary to calculate the squish velocity from the deflection of the thin plate of detector. It is assumed that the state change of gas in the cylinder is polytropic and the compression begins at the position where the piston crown closes the ports. With the detector fitted to the cylinder head, the relation between the deflection of thin plate and the velocity

Fig. 2 Electronic circuit for velocity measurement

is calibrated in the atmospheric surrounding. Using the velocity u_0 , which gives the deflection equal to the one measured in the cylinder, the squish velocity u is written as follows:

$$u = \sqrt{\gamma_0 / \gamma_c} (p_c / p)^{1/2m} u_0 \quad \dots\dots\dots (1)$$

The pressure p_c at the beginning of compression and the polytropic exponent m are obtained from the pressure change measured with a piezo pressure transducer. The specific weight of gas γ_c at the beginning of compression is calculated from the pressure p_c and the gas temperature measured with a fine platinum wire of the diameter of 0.02 mm. The wire temperature at the piston position near the bottom dead center is estimated to follow the gas temperature in the cylinder, because of the slow change of gas temperature. Therefore the gas temperature at the beginning of compression is assumed to be equal to the wire temperature.

Figure 3 shows the velocity detector fixed to the cylinder head. The thin steel plate of detector is positioned 1.5 mm apart from the side wall of combustion chamber, and its face is made perpendicular to the direction of cylinder radius, that is, the direction of squish velocity. The thin plate is intended to ensure that only the radial component

Fig. 3 Sectional view of apparatus

Fig. 4 Sectional view of combustion chambers

of velocity is detected, the other components having no effect upon the deflection. The position of detector can be moved in the axial direction of cylinder, using a packing of proper thickness.

Sectional views of combustion chambers used are shown in Fig. 4. The diameter ratio of the chamber to the cylinder is 0.45 and 0.55. The volumes of those combustion chambers are designed so that the geometric compression ratio is 16 for the top clearance of 1 mm. The squish velocity has been

Position: Center position of pressure sensitive face measured from cylinder head
d/D=0.45 $\Delta s=1.0$ mm

Fig. 5 An example of deflection of the thin plate recorded

measured at the motoring speed of 1500 rpm.

4. Experimental results and considerations

Fitting the velocity detector to the cylinder head, the deflection of the thin plate due to the squish velocity has been measured. An example of the deflection change recorded is shown in Fig. 5. Since the magnitude of deflection varies considerably from cycle to cycle, the samples near the average are picked up in the figure. As the detector is small, it is very difficult to make the compensating solenoid which compensates completely for the change of magnetic permeability of gas. Therefore the measurement has been carried out by the following procedure. After measuring the deflection, the thin plate of detector is fixed so as not to be deflected by the gas velocity. Then the apparent deflection occurring due to the factors other than the velocity is measured in the same condition of engine running. The measured deflection is corrected by subtracting the apparent one.

The magnitude of squish velocity at a given crank angle varies considerably. The range of variation is within the hatched domain in Fig. 6. While the variation is considered to be owing to the measuring error, it is estimated that the velocity itself varies from cycle to cycle. In this experiment the squish velocity is defined as the average value calculated from the deflection changes for 50 cycles, which are recorded with a long recording camera attached to the synchroscope. At the position of piston where the center of the pressure sensitive face of thin plate comes inside the combustion chamber, the measured velocity becomes small and consequently does not give the squish velocity. In Fig. 6 the velocity in the range of crank angles smaller than 15 deg. before t.d.c. does not indicate the true squish velocity.

Fig. 6 Range of variation of squish velocity

Consider the part of the pressure sensitive face of thin plate which juts out into the cylinder from the cylinder head. Taking the center of this part as the measuring position, the distribution curve of squish velocity between the cylinder head and the piston crown can be obtained for a given crank angle. The distribution curves show a tendency that the squish velocity becomes a little larger with approaching of the piston crown. The squish velocity at the piston crown can be estimated by extending the curve. The results obtained are shown in Fig. 7. The squish velocity becomes maximum at 5~10 deg. before t.d.c. The theoretical velocity described later shows the same tendency as the experimental one. And the magnitude of velocity calculated under the condition of no loss is not much different from the experimental value.

5. Theoretical considerations

5.1 Theoretical squish velocity

As the piston approaches the top dead center, the radial movement of gas occurs in the annular space between the piston crown and the cylinder head which is shown as the hatched domain of Fig. 8. Assuming that the cylinder pressure is spatially uniform during compression, the radial velocity at the side wall of combustion chamber, that is, the theoretical squish velocity is given by the following equation⁽⁹⁾.

Fig. 7 Magnitude of squish velocity

Fig. 8 Model of gas flow into combustion chamber

$$\frac{u_{th}}{c_m} = -\frac{\pi}{4} \frac{1-(d/D)^2}{(s/D)(d/D)} \times \frac{\varepsilon_2}{\{\varepsilon_1 + \varepsilon_2 + h(\theta)\} \{\varepsilon_1 + h(\theta)\}} \frac{dh(\theta)}{d\theta} \dots\dots\dots (2)$$

where $\varepsilon_1 = \Delta s/s$, $\varepsilon_2 = V_k/V_h$, and

$$h(\theta) = \frac{1}{2} \left\{ 1 - \cos \theta + \frac{1}{\lambda} (1 - \sqrt{1 - \lambda^2 \sin^2 \theta}) \right\}$$

Using the engine sizes of $D=85$ mm, $s=93$ mm, $1/\lambda=3.76$ and $\varepsilon_2=0.056$, the theoretical squish velocity was calculated in non-dimensional form. The results obtained are shown in Fig. 9 and also by the solid lines in Fig. 7. It is clearly known from Fig. 9 that the theoretical squish velocity becomes larger with the decrease in the diameter ratio d/D of the combustion chamber to the cylinder or the top clearance Δs . The theoretical velocity is relatively small in the range of crank angles larger than 30 deg. before t.d.c. It increases as the piston approaches the top dead center. After reaching a maximum, it decreases and becomes zero at the top dead center. The relation between the maximum value of the velocity curve and the diameter ratio d/D is shown in Fig. 10.

There are many factors causing a decrease of squish velocity such as viscosity of gas, friction, heat loss, gas leakage, etc. In this paper, only the gas leakage through the piston rings and the heat loss to the wall are theoretically investigated, because their effects on the squish velocity are estimated to be more remarkable than the other factors. The heat loss and the gas leakage reduce always the squish velocity, and the velocity decrement becomes larger as the top dead center is approached.

5.2 Decrement of squish velocity due to the gas leakage through the piston rings

The outward radial velocity occurs in the cylinder when gas leaks through the piston rings during compression stroke. Therefore, the squish velocity is reduced by the gas leakage. The decrement of squish velocity Δu_r is calculated on the following assumptions.

(1) Since gas leaks uniformly throughout the

Fig. 9 Theoretical squish velocity

piston rings, the velocity decrement at the side wall of combustion chamber is spatially uniform. Flow through the piston rings can be replaced by that through the equivalent hole, whose effective flow area is kept constant during compression stroke.

(2) The gas in the cylinder leaks through the equivalent hole under the condition of critical state, and its state change through the hole is adiabatic.

(3) The pressure and temperature in the cylinder, which are spatially uniform, change polytropically with a constant exponent. The variation of polytropic exponent due to the gas leakage can be ignored.

The flow rate of gas into the combustion chamber, which occurs as the result of squish, is reduced by the gas leakage. The decrement of the instantaneous flow rate at the cylindrical face involving the side wall of combustion chamber is equal to the leaking rate of gas multiplied by the ratio of the volume inside the cylindrical face to the cylinder volume. Therefore the instantaneous leaking rate dG_r/dt is connected with the velocity decrement Δu_r by the following equation.

$$\pi d s \{\varepsilon_1 + h(\theta)\} \Delta u_r \gamma = \frac{(d/D)^2 \{\varepsilon_1 + h(\theta)\} + \varepsilon_2}{\varepsilon_1 + \varepsilon_2 + h(\theta)} \frac{dG_r}{dt} \dots\dots\dots (3)$$

Since the cylinder pressure over the critical value is assumed, the instantaneous leaking rate dG_r/dt is

$$\begin{aligned} \frac{dG_r}{dt} &= \mu f \left(\frac{2}{\kappa+1} \right)^{\{\kappa+1/2(\kappa-1)\}} \sqrt{\kappa g p \gamma} \\ &= \mu f \left(\frac{2}{\kappa+1} \right)^{\{\kappa+1/2(\kappa-1)\}} \\ &\quad \times \sqrt{\kappa g p_c \gamma_c} \left\{ \frac{\varepsilon_1 + \varepsilon_2 + h(\theta_c)}{\varepsilon_1 + \varepsilon_2 + h(\theta)} \right\}^{(m+1)/2} \dots\dots\dots (4) \end{aligned}$$

Eliminating dG_r/dt from Eqs. (3) and (4), the non-dimensional decrement $\Delta u_r/c_m$ is represented by

Fig. 10 Relation between maximum value of theoretical squish velocity and diameter ratio d/D of combustion chamber to cylinder

$$\frac{\Delta u_r}{c_m} = \frac{1}{8} \left(\frac{2}{\kappa+1} \right)^{(\kappa+1/2)(\kappa-1)} \frac{\{\epsilon_1 + \epsilon_2 + h(\theta_c)\}^{(m-1)/2}}{(d/D)(s/D)} \times \frac{60\mu f \sqrt{\kappa g R T_c}}{n V_h} \times \frac{(d/D)^2 \{\epsilon_1 + h(\theta)\} + \epsilon_2}{\{\epsilon_1 + h(\theta)\} \{\epsilon_1 + \epsilon_2 + h(\theta)\}^{(m+1)/2}} \dots (5)$$

Effects of the engine speed n , the stroke volume V_h and the effective leaking area μf can be synthetically investigated by using the non-dimensional number of $60\mu f \sqrt{\kappa g R T_c} / n V_h$. Assuming that compression begins at the bottom dead center, and using the dimensions of experimental engine and $m=1.4$, the decrement of squish velocity is calculated from Eq. (5). The results obtained are shown in Fig. 11. The velocity decrement $\Delta u_r / c_m$ becomes larger as the top dead center is approached, because of both the increase in the cylinder pressure and the decrease in the clearance between the piston crown and the cylinder head. Maximum velocity decrement is attained at the top dead center. Figure 12 shows the maximum decrement calculated for $m=1.4$ and 1.2. The velocity decrement $\Delta u_r / c_m$ is proportional to the non-dimensional number of

$$60\mu f \sqrt{\kappa g R T_c} / n V_h,$$

and increases with the decrease in the diameter ratio d/D of the combustion chamber to the cylinder. It decreases also, when the polytropic exponent m becomes smaller, due to the lower compression pressure.

Before rebuilding the engine for the squish measurement, the volume of leakage gas through the piston rings has been measured in a motoring state of the engine. The leakage volume per cycle is about 1.6% of the stroke volume at 1500 rpm.

Fig. 11 Decrement of squish velocity due to gas leakage through piston ring

Using the measured polytropic exponent and the measured pressure at the compression beginning, and furthermore assuming a gas temperature of 298°K at the beginning, the effective leaking area of piston rings which produces the same leaking volume as the measured one is calculated by integrating Eq. (4) for the period of compression and expansion strokes. The value of $60\mu f \sqrt{\kappa g R T_c} / n V_h$ which gives a leakage gas volume of 1.6% at 1500 rpm is 0.0065. Even if the gas volume leakage during the compression stroke is 2.5% of the stroke volume, the value of the above non-dimensional number is 0.02. Therefore, the velocity decrement due to the gas leakage does not seem to be large in actual engines. The ratio of the velocity decrement Δu_r to the theoretical squish velocity u_{th} is shown in the upper part of Fig. 11; $\Delta u_r / u_{th}$ increases as the top dead center is approached. Although it is infinity at top dead center, it is not large in the range of 5~10 deg. before t.d.c. where the theoretical squish velocity becomes maximum. Therefore it is evident that the gas leakage through the piston rings does not remarkably affect the squish velocity. Comparing the curve of $60\mu f \sqrt{\kappa g R T_c} / n V_h = 0.01$ and $d/D = 0.45$ in Fig. 11 with that of $d/D = 0.55$, it is known that the ratio $\Delta u_r / u_{th}$ becomes smaller with the decrease in the diameter ratio of the combustion chamber to the cylinder.

5.3 Decrement of squish velocity due to the heat loss to the wall

Whenever the gas temperature in the cylinder exceeds the wall temperature, heat flow occurs from the gas to the wall. Since the gas outside the cylindrical face involving the side wall of combustion chamber, that is, the outside gas is cooled more strongly than the inside gas, the velocity with the outward radial component is produced in the cylinder owing to the heat loss. This velocity becomes larger as the piston approaches the top dead center.

Fig. 12 Relation between non-dimensional number of $60\mu f \sqrt{\kappa g R T_c} / n V_h$ and maximum decrement of squish velocity

As the heat flow process in engines is very much complicated, it is difficult to find the wall temperature distributions of the cylinder, the cylinder head and the piston crown, and also to know the heat transfer rate in every part of the wall. Therefore the decrement of squish velocity due to the heat loss is calculated on the following simple assumptions.

(1) During the compression stroke, the outside gas in the space between the cylinder head and the piston crown, which is indicated by the hatched domain in Fig. 8, and the inside gas are cooled, respectively, by only the wall which is touched by each gas.

(2) The wall temperature is uniform and is kept constant during the compression stroke.

(3) The heat transfer rate of wall is uniform throughout the wall.

(4) The mean state of gas in the cylinder changes polytropically with a constant exponent.

The equation of energy for the outside gas shown as the hatched domain in Fig. 8 is

$$\frac{d(c_v T G_a)}{dt} = -\frac{F_a}{F} \frac{dQ}{dt} - A p \frac{dV_a}{dt} + c_p T \frac{dG_a}{dt} \quad \dots\dots\dots(6)$$

The relation between the squish velocity u and the weight of outside gas G_a is represented by

$$\frac{dG_a}{dt} = -\pi d s \{\epsilon_1 + h(\theta)\} \frac{p}{RT} u \quad \dots\dots\dots(7)$$

Eliminating dG_a/dt from the above two equations, we obtain

$$u = -\frac{\kappa - 1}{\kappa \pi d s \{\epsilon_1 + h(\theta)\} p} \left\{ \frac{1}{A} \frac{d(c_v T G_a)}{dt} + \frac{1}{A} \frac{F_a}{F} \frac{dQ}{dt} + p \frac{dV_a}{dt} \right\} \quad \dots\dots\dots(8)$$

When the cooling of gas in the cylinder is spatially uniform, the decrement of squish velocity due to the heat loss is not produced. Therefore, replacing $(F_a/F)(dQ/dt)$ with $(V_a/V)(dQ/dt)$ to

Fig. 13 Heat transfer rate

satisfy the condition of uniform cooling, Eq. (8) gives the theoretical squish velocity u_{th} . Ignoring the differences of temperature and pressure between the uniform and non-uniform coolings, the decrement of squish velocity $\Delta u_h = u_{th} - u$ is given by

$$\begin{aligned} \frac{\Delta u_h}{c_m} &= \frac{(\kappa - 1) 30}{\kappa \pi d s^2 n \{\epsilon_1 + h(\theta)\} p} \left(\frac{F_a}{F} - \frac{V_a}{V} \right) \frac{1}{A} \frac{dQ}{dt} \\ &= \frac{\kappa - 1}{\kappa} \frac{30}{\pi d s^2 n p_c} \left\{ \frac{\epsilon_1 + \epsilon_2 + h(\theta)}{\epsilon_1 + \epsilon_2 + h(\theta_c)} \right\}^m \frac{1}{\epsilon_1 + h(\theta)} \\ &\times \left[\frac{F_a}{F} - \frac{\{1 - (d/D)^2\} \{\epsilon_1 + h(\theta)\}}{\epsilon_1 + \epsilon_2 + h(\theta)} \right] \frac{1}{A} \frac{dQ}{dt} \quad \dots\dots\dots(9) \end{aligned}$$

The rate of heat loss dQ/dt is

$$\frac{dQ}{dt} = \alpha \left[T_c \left\{ \frac{\epsilon_1 + \epsilon_2 + h(\theta_c)}{\epsilon_1 + \epsilon_2 + h(\theta)} \right\}^{m-1} - T_w \right] F \quad \dots\dots\dots(10)$$

There are many empirical formulas for the heat transfer rate of engine. The following formula presented by Woschni⁽¹⁰⁾ is used in this paper.

$$\alpha = 265 D^{-0.214} (c_m p)^{0.786} T^{-0.525} \quad \dots\dots\dots(11)$$

(α : kcal/m²hr°C, D : m, c_m : m/sec, p : kg/cm², T : °K)

It is not adequate to calculate the rate of heat loss from the heat release equation for polytropic change, because the rate of heat loss and consequently the decrement of squish velocity due to the heat loss become zero at top dead center. Actually the rate of heat loss increases as the top dead center is approached. Therefore the rate of heat loss is calculated from Eq. (10) on the assumption that the pressure and temperature in the cylinder change polytropically. The wall temperature T_w in Eq. (10) is set from the following equation, so that the total heat loss is equal to the total heat release in the polytropic change.

Fig. 14 Decrement of squish velocity due to heat loss to wall

$$\frac{(\kappa-m)A_p V_h}{(\kappa-1)(m-1)} \left[\frac{(\varepsilon_1 + \varepsilon_2 + h(\theta_c))^m}{(\varepsilon_1 + \varepsilon_2)^{m-1}} - \{\varepsilon_1 + \varepsilon_2 + h(\theta_c)\} \right] = \frac{1}{120\pi n} \int_{\theta_c}^0 \alpha \times \left[\left\{ \frac{\varepsilon_1 + \varepsilon_2 + h(\theta_c)}{\varepsilon_1 + \varepsilon_2 + h(\theta)} \right\}^{m-1} - \frac{T_w}{T_c} \right] T_c F d\theta \dots\dots\dots (12)$$

It is assumed that the compression begins at bottom dead center, the pressure and the temperature at the compression beginning are 1.033 ata and 350°K respectively, and the polytropic exponent is 1.35. Using the dimensions of experimental engine, the rate of heat transfer and the decrement of squish velocity due to the heat loss are calculated at the engine speed of 1500 rpm. The results obtained are shown in Figs. 13 and 14. The calculated value of wall temperature amounts to 260 ~ 300°K, being lower than the actual temperature of wall estimated. It is possible to make the wall temperature reasonable by modifying the polytropic exponent. However, the authors suppose that the rate of heat transfer calculated from Eq. (11) is lower than the actual one. The low wall temperature is not corrected in this theoretical calculation from the point of view that the error due to the low rate of heat transfer is compensated by the low wall temperature.

As shown in Fig. 14, the decrement of squish velocity is small during the early stage of compression stroke, becoming remarkable at crank angles smaller than 20 deg. before t.d.c. It increases as the top dead center is approached, reaching a maximum at the top dead center. The increase in the velocity decrement is owing to the following facts: both the heat transfer rate and the temperature difference between the gas and the wall increase, and moreover the difference of cooling rate between the outside and inside gases becomes larger due to the decrease in the clearance between the piston crown and the cylinder head. The velocity decrement increases with a decrease in the diameter ratio d/D of the combustion chamber to the cylinder or the top clearance Δs . However, the ratio $\Delta u_h/u_{th}$ of the decrement to the theoretical squish velocity is scarcely affected by d/D as evident from the comparison of the curve of $d/D=0.45$ in the upper part of Fig. 14 with that of $d/D=0.55$; $\Delta u_h/u_{th}$ increases as the piston approaches the top dead center. However, the heat loss to the wall is estimated to have virtually little effect on the squish velocity, taking the fact into consideration that the theoretical squish velocity becomes a maximum at 5 ~ 10 deg. before t.d.c., being zero at the top dead center. Although the process of heat loss assumed differs from that in engines, the results obtained might be considered to display qualitatively the influence of heat loss on the squish.

Dotted lines in Fig. 7 indicate the squish velocity with the heat loss and the gas leakage, which is calculated under the same condition as that in the experiment. Since the value of polytropic exponent is small, being in the range of 1.23 ~ 1.25, the velocity decrement due to the heat loss is relatively large. Therefore, the difference between the dotted line and the solid line, which is the theoretical squish velocity, is mainly owing to the heat loss. The velocity decrement due to the heat loss becomes smaller with an increase in the polytropic exponent. Comparing the measured velocity in Fig. 7 with the calculated one, it may be estimated that the velocity produced in actual engines is close to the theoretical squish velocity.

6. Conclusions

By motoring a model engine, the radial gas velocity at the brim of combustion chamber, which produces a toroidal motion in the chamber, that is, the squish velocity has been measured. Furthermore the heat loss to the wall and the gas leakage through the piston rings, which have supposedly a relatively large influence on the squish velocity, have been theoretically investigated on simple assumptions. The measured squish velocity is close to the ideal one calculated under the conditions without losses. The velocity produced in actual engines may be not much different from the theoretical squish velocity. The velocity decrement due to the heat loss and the gas leakage is relatively large when the crank angle is nearer to the top dead center than 20 deg. before t.d.c., becoming maximum at the top dead center. However, the absolute value of decrement is too small to affect remarkably the squish velocity.

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