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Thermal Effect upon Air Capacity of the Four-Stroke Engine

(1st Report, Thermodynamical Analysis Method)

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The temperature changes of the mixture gas in the induction process are given using the following notation: ΔT_1 : Vaporization of fuel, ΔT_2 : Ram effect, ΔT_3 : Preheating for intake mixture, ΔT_4 : Heat transfer from high temperature parts, ΔT_5 : Mixing with residual gas. The change ΔT_5 has almost no effect on air capacity, but ΔT_2 , ΔT_3 , and ΔT_4 change with engine types, operation and atmospheric conditions.

Report No. 1 has expressed the sum of ΔT_2 , ΔT_3 , and ΔT_4 (called $\Delta T_{2,3,4}$) as a function of the directly measurable intake mixture gas weight and the cylinder pressures. The suction end temperature T_1 is a function of $\Delta T_{2,3,4}$ and pressures needed for thermodynamical analysis. It has become possible to determine each of the changes ΔT_1 to ΔT_5 analytically. Finally the relationship between the air capacity and various temperature rises of the mixture gas is expressed theoretically.

1. Introduction

Qualitatively, the effects of the pressure of air intake, the pressure of residual gas, the temperature of air intake, compression ratio, the time of valve opening and that of valve closing, the length of the intake pipe and exhaust pipe, the cylinder wall temperature, etc., on G_a , the weight of the air intake, have already been clarified. It is thought that in the revised formula of developed power relating to G_a , the developed power is generally in inverse proportion to the temperature of air intake. However, any satisfactory formula which enables us to estimate quantitatively the changes in G_a in different types of engines, under different operating and atmospheric conditions has not yet been found. The major reason is that in the studies undertaken so far, G_a was expressed only as a function of T_1 , the temperature at the end of suction, and T_5 , the temperature of residual gas⁽¹⁾. In other words, because it is very difficult to measure T_1 and T_5 , estimated values were used. However, if the mixture of high-temperature residual gas and low-temperature intake mixture is considered, it is understood that T_5 , the temperature of the residual gas, has almost no effect on G_a ⁽²⁾. Namely, if it is

assumed that the two gases mix and the temperature difference is equalized, the high-temperature residual gas shrinks as a result of cooling while the low-temperature intake mixture expands as a result of warming. Therefore, the changes in the volumes of the two gases cancel each other and scarcely affect G_a . After all, G_a is not related to T_5 . The prevailing thinking that G_a is in inverse proportion to T_1 , the average temperature at the end of suction, is wrong. It should be assumed that G_a is in inverse proportion to $(T_1 - \Delta T_5)$ where (ΔT_5) is a rise in the temperature due to the mixing of residual gas.

The purpose of this first report is to explain the method to find the amount of temperature change in the mixture gas produced during suction and the temperature at the end of suction by using a formula obtained from thermodynamic analysis of the cycle and G_s , the value of the intake mixture obtained experimentally and the value of the pressure.

In this report the author will also propose a new formula to express G_a .

2. A method of thermodynamic analysis for measurement of gas temperature

Several methods to measure the temperature of the gas in a given cylinder have been tested in the past, including an optical method⁽³⁾, a sonic method⁽⁴⁾ and a method using thermoelectric couples⁽⁵⁾.

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Recently a method which uses high frequency⁽⁶⁾ has been proposed. But all have a common drawback: Only the temperature of that part of the gas which is in the passage of the rays or sound waves, or which borders on the measured point can be measured but the average temperature of the entire gas cannot be measured. Moreover, mounting an instrument to measure the average temperature of the entire gas in a real engine poses fairly difficult structural problems which can easily hamper the proper functioning of the engine.

The method to measure the temperature of gas proposed here is to measure the gas temperature T by measuring the pressure P and the gas weight G and using the equation $PV=GRT$ when the values of the cylinder capacity V and the gas constant R are known.

This method is based on the principle of the gas thermometer. Therefore, by this method, one can find the average temperature of the gas in a cylinder without difficulty from the gas pressure in the cylinder and the weight of the intake mixture both of which are relatively easy to measure in a real engine.

2.1 The thermal change in the intake mixture during suction and the method to measure the temperature of the gas at the end of suction⁽⁷⁾

[A thermodynamical method of analysis]

Because a new air intake is considered to receive the thermal effects of the aforementioned five changes of ΔT_1 to ΔT_5 by the end of suction, we have

$$T_1 = T_a - \Delta T_1 + \Delta T_{2,3,4} + \Delta T_5 \dots \dots \dots (1)$$

if for the sake of simplicity we express the intermediate thermal changes as

$$\Delta T_2 + \Delta T_3 + \Delta T_4 = \Delta T_{2,3,4}$$

Because $T_1 - \Delta T_5 = T_a - \Delta T_1 + \Delta T_{2,3,4}$ we can also find the temperature as $(T_a - \Delta T_1 + \Delta T_{2,3,4})$ instead of $(T_1 - \Delta T_5)$ as the effect of temperature on G_a , as explained in the foreward. Here, T_a can be measured by thermometer, and ΔT_1 can be theoretically calculated from the latent heat of vaporization of the fuel and the air-fuel ratio but the value of $\Delta T_{2,3,4}$ should differ with the type of engine, the operating and atmospheric conditions. But values of

$\Delta T_{2,3,4}$ have never been actually measured and a few estimated values⁽⁸⁾ which have been announced stand on a very doubtful logical basis.

In finding the value of $\Delta T_{2,3,4}$ the P - V and T - V diagram at the time of operating a four-cycle gasoline engine, with the throttle fully open but without supercharging, is idealized and conceived as Fig. 1, and the following assumptions are made to simplify the analysis:

- (1) The overlapping angle of a valve at the top dead center is zero.
- (2) The pressures in the intake and exhaust pipes during suction and exhaust strokes, and the pressure in the cylinder are constant.
- (3) The drop in the pressure which occurs when gas passes the intake valve is instantaneous at the top dead center.

Notations

T_a	: the temperature of the intake mixture ($^{\circ}\text{K}$)
T_1	: the average temperature of the gas mixture at the end of suction
ΔT_1	: a decline in the temperature due to evaporation of fuel
ΔT_2	: a rise in the temperature due to mechanical effects such as inertia effects ($^{\circ}\text{C}$); (in throttled driving, $\Delta T_2 \approx 0$)
ΔT_3	: a rise in the temperature due to the preheating of intake ($^{\circ}\text{C}$) ($\Delta T_3 \approx 0$ in an engine without a preheater)
ΔT_4	: a rise in the temperature due to the transmission of heat from high-temperature parts of the cylinder wall and the top of the piston ($^{\circ}\text{C}$)
ΔT_5	: a rise in the temperature due to the mixing of intake mixture and residual gas ($^{\circ}\text{C}$)
R_1	: the gas constant of the entire gas in the cylinder at the end of suction
R_5	: the gas constant of burning gas
n_1	: the polytropic index during exhaust (Figure 1, 4~5)
n_2	: the polytropic index during the first half of intake (Fig. 1, 5~6)
V_s	: the effective cubic capacity of the stroke with the angle of a closed intake valve considered
V_c	: the cubic capacity of the void
ϵ	: the effective compression ratio with the closure of the intake valve considered
G_a	: the weight of air intake per cycle
G_f	: the weight of fuel per cycle
G_s	: the weight of intake mixture per cycle
G_r	: the weight of residual gas per cycle
Y	: the air-fuel ratio $= G_a/G_f$
c_{ps}	: the average specific heat of the intake mixture at a constant pressure
c_{pr}	: the average specific heat of the residual gas at a constant pressure.
The notations for the various points in a cycle (Fig. 1)	
1	: The end of suction (when the intake valve is closed)
2	: The end of compression (top dead center)
3	: The end of combustion (top dead center)
4	: The end of inflation (when the exhaust valve is closed)
5	: The end of exhaust (top dead center)
6	: The point where the pressure in the intake pipe and the pressure of residual gas are equal
7	: The point where the inflated residual gas has been totally absorbed in the cylinder
7-8	: The process of suction at a constant pressure
8-1	: The process of a rise in the pressure due to the inertia of intake
P_s	: The pressure in the intake pipe
P_e	: The pressure in the exhaust pipe
P_1	: The pressure in the cylinder during a suction stroke

The subscripts attached to P and T shows the pressure and temperature at each point.

Fig. 1 The P - V plot and T - V plot of an ideal cycle

(4) A rise in the pressure due to the inertia of the intake is instantaneous at the bottom dead center.

Condition 5 in Fig. 1 is the condition after the gas was inflated from P_4 to P_5 . Therefore,

$$T_5 = T_4 \left(\frac{P_5}{P_4} \right)^{(n_1-1)/n_1} \dots\dots\dots (2)$$

Because $P_5 > P_i$ in the case shown in Fig. 1, if the intake valve opens at the top dead center, a portion of the residual gas flows back into the intake pipe, expanding to P_i in its reflux. In the meantime, the piston comes down and the inside capacity of the cylinder increases slightly, and at one point, the pressure of the residual gas becomes equal to P_i . Later, the residual gas which has flowed back into the intake pipe is totally absorbed in the cylinder. If we ignore the distance of lowering of the piston because it is negligibly short, these points can be indicated by point 6 and point 7, and this process can be expressed as

$$V_7 = V_c \left(\frac{P_5}{P_7} \right)^{1/n_2}$$

But in throttled operation, $P_7 = P_8 = P_1$. In case of inertia effect, $\Delta T_2 = T_1 - T_8$, which is the rise in the temperature due to this effect, reaches its maximum value at about 5°C as mentioned in Section 2.3 and is fairly small as compared with other thermal changes. Therefore the pressure at the time of considering the temperature of the residual gas at Point 7, where the new intake actually starts, may be assumed as $P_7 = P_1$. Accordingly,

$$T_7 = T_5 \left(\frac{P_1}{P_5} \right)^{(n_2-1)/n_2} \dots\dots\dots (3)$$

The cylinder capacity is nearly the same for condition 1 and condition 4. Accordingly, it may be assumed that the following relationship applies approximately*:

$$T_1 = T_4 \frac{P_1}{P_4} \frac{R_5}{R_1} \dots\dots\dots (4)$$

If T_4 is eliminated from Eq. (2) and Eq. (4), we have

$$T_5 = T_1 \frac{R_1}{R_5} \frac{P_4}{P_1} \left(\frac{P_5}{P_4} \right)^{(n_1-1)/n_1} \dots\dots\dots (5)$$

Strictly speaking, the pressure during suction 7~8 is not constant. But the piston speed is generally proportional to the open area of the intake valve at each crank angle during suction. Accordingly, the speed at which the new air intake passes through

the intake valve is nearly constant, and the decrease of the pressure due to the throttling of the valve is nearly constant during suction.**

Assuming that mixing of the intake mixture and the residual gas in the cylinder during a suction stroke occurs under a constant pressure, the heat which the intake mixture absorbs and the heat which the residual gas loses equal each other. Accordingly, we have the following approximation:***

$$\Delta T_5 c_{ps} G_s = (T_7 - T_1) c_{pr} G_5 \dots\dots\dots (6)$$

where $\Delta T_5 = \{T_1 - (T_a - \Delta T_1 + \Delta T_{2,3,4})\}$

From the relationship $V_c = V_s / (\epsilon - 1)$ the weight of the residual gas, represented as G_5 , is

$$G_5 = \frac{V_s}{\epsilon - 1} \frac{P_5}{R_5 T_5} \dots\dots\dots (7)$$

V_1 , which is the capacity of the cylinder at the end of suction, can be expressed as $V_1 = \{\epsilon / (\epsilon - 1)\} V_s$. Therefore, if the total gas weight in the cylinder is expressed as G_1 , the weight of the new intake mixture, shown as G_s , is:

$$G_s = G_1 - G_5 = V_s \left\{ \frac{P_1}{R_1 T_1} \frac{\epsilon}{\epsilon - 1} - \frac{P_5}{R_5 T_5} \frac{1}{\epsilon - 1} \right\} \dots\dots\dots (8)$$

If T_5 and T_1 are eliminated from Eq. (7) and Eq. (8) by using Eq. (5), we have

$$G_s = G_s \left\{ \frac{1}{1 - \frac{1}{\epsilon} \left(\frac{P_5}{P_4} \right)^{1/n_1}} - 1 \right\} \dots\dots\dots (7)'$$

If the value of the polytropic index n_1 is known, the value of G_5 can be indirectly found from P_5 , P_4 , and the measured value of G_s . If G_5 of Eq. (7) and G_s of Eq. (8) are substituted into Eq. (6) and T_5 is eliminated from Eq. (3) and Eq. (5), T_1 , the temperature at the end of suction, is expressed as

$$T_1 = (T_a - \Delta T_1 + \Delta T_{2,3,4}) \frac{1 - \frac{1}{\epsilon} \left(\frac{P_5}{P_4} \right)^{1/n_1}}{1 - \frac{1}{\epsilon} \left\{ \frac{R_1}{R_5} \frac{c_{pr}}{c_{ps}} \left(\frac{P_5}{P_1} \right)^{1/n_2} + \left(\frac{P_5}{P_4} \right)^{1/n_1} \left(1 - \frac{c_{pr}}{c_{ps}} \right) \right\}} \dots\dots\dots (9)$$

If G_5 is eliminated from Eq. (6) and Eq. (7) and T_5 is eliminated by substituting the result into Eq. (8) and if $\Delta T_{2,3,4}$ is found by eliminating T_1 from Eq. (1), we have

$$\Delta T_{2,3,4} = \frac{P_1 V_s}{G_s R_1} \frac{\epsilon}{\epsilon - 1} \left\{ 1 - \frac{1}{\epsilon} \left(\frac{P_5}{P_1} \right)^{1/n_2} \frac{R_1}{R_5} \right\} - T_a + \Delta T_1 - \left(1 - \frac{c_{ps}}{c_{pr}} \right) \Delta T_5 \dots\dots\dots (10)$$

Namely, T_1 of Eq. (9), as a function of the temperature, includes $\Delta T_{2,3,4}$ and $\Delta T_{2,3,4}$ of Eq. (10) contains ΔT_5 .

However, if the intake mixture and the residual gas are assumed to be the same gas and the relations $C_{ps} = C_{pr}$ and $R_1 = R_5$ are assumed, Eq. (9) will be

* When there is gas leak from the void of the piston, the relationship expressed in Eq. (4) does not hold true. But the gas leak is about 0.5% of the air intake at its maximum⁽⁹⁾. Therefore, the author proceeds with the discussion here on the assumption that Eq. (4) is valid.

** A concrete example will be shown in Fig. 4.

*** When there is no inertia effect, Eq. (6) applies completely.

$$T_1 = (T_a - \Delta T_1 + \Delta T_{2,3,4}) \frac{1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_4} \right)^{1/n_1}}{1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n_2}} \dots\dots(11)$$

and Eq. (10) can be expressed as

$$\Delta T_{2,3,4} = \frac{P_1 V_5}{G_s R_1} \frac{\varepsilon}{\varepsilon - 1} \left\{ 1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n_2} \right\} - T_a + \Delta T_1 \dots\dots(12)$$

Eq. (12) which represents the approximations of $\Delta T_{2,3,4}$ does not include ΔT_5 . If the precise equation and the approximate equation which present T_1 and $\Delta T_{2,3,4}$ are compared, the average specific heat ratio⁽¹⁰⁾ and the gas constant ratio⁽⁷⁾ between the intake mixture and the residual gas, respectively, may be considered as

$$\frac{c_{ps}(60^\circ\text{C})}{c_{pr}(60^\circ\text{C})} = \frac{0.25}{0.28} \doteq 0.90, \quad \frac{R_1}{R_5} = \frac{28.0}{29.0} \doteq 0.93$$

within the range of normal air-fuel ratio. Because the value of ΔT_5 is 30~50°C in unthrottled operation and about 100°C in idling, Eq. (2) indicates the value of T_1 to be about 2% higher than does Eq. (11), which is an approximation. But an error of this degree seems to be negligible for the measured value of a temperature if the errors in measuring the amount of the air intake, pressure, etc., are considered. Therefore, the author will use the approximate equation (11) and approximate equation (12) in the actual measurement of temperatures.

In a real engine, the opening angle of the exhaust valve and the closing angle of the intake valve are almost the same with the bottom dead center as the center. Accordingly, it may be considered that the cylinder capacity of 1 and that of 4 in the P - V diagram are nearly the same. If the effective compression ratio is determined in this way, a small error in measurement arises because of the dynamic and/or thermal effects of the valve assembly and, even if the effective compression ratio is estimated smaller than it really is, the value of P_1 in the indicator diagram is presented larger than real. Accordingly, the error has almost no effect on the values of $\Delta T_{2,3,4}$ of Eq. (12).

Because G_s , the weight of mixture gas, is $G_a + G_f$, the value of G_s can be found by obtaining the values of G_a and G_f by experiment. The pressures P_1 , P_4 and P_5 can be found from the low-pressure or medium-pressure indicator diagram, and T_a , the temperature of air intake, can be measured with a thermometer.

ΔT_1 , which represents a drop in the temperature of the fuel due to vaporization, can be calculated from the latent heat of vaporization of the fuel and the air-fuel ratio. R_1 , which is the gas constant of the entire gas at the end of suction, can be calculated from the air-fuel ratio and the mixing ratio of the residual exhaust gas. The values of the

polytropic indices n_1 and n_2 may be assumed to be $n_1 = n_2 = 1.30$

as will be discussed in detail in Section 3.2.1 and section 3.2.2 of this report. By substituting these values in Eq. (12), the values of $\Delta T_{2,3,4}$ are found, and by substituting the values of $\Delta T_{2,3,4}$ in Formula (11), T_1 , which is the temperature at the end of suction, is determined. Hereinafter the author will present concrete methods to separate and find $\Delta T_1 \sim \Delta T_5$, which are changes in the temperature related to the aforementioned values.

2.2 A decline in the temperature of the fuel due to vaporization (ΔT_1)

(a) In the case where the entire fuel vaporizes

If we assume that the fuel sucked in by the end of suction is totally gasified, T_{10} , which is the decline in the temperature, is expressed as Eq. (13) because the latent heat of vaporization of the fuel is absorbed by the intake air and the gasified fuel. If the decline in the temperature in that process is represented by ΔT_{10} , we have

$$\Delta T_{10} = L / (Y c_{pa} + c_{pf}) \dots\dots\dots(13)$$

If the fuel is not completely gasified, we have

$$\Delta T_1 = x \Delta T_{10} \dots\dots\dots(14)$$

where x represents the percentage of the gasified fuel in the total fuel (hereinafter referred to as the vaporization percentage).

Here, c_{pa} is the specific heat of air at constant pressure and is 0.24 kcal/kg°C;

c_{pf} is the specific heat of fuel vapor at constant pressure and is 0.369 kcal/kg°C;

L is the latent heat of vaporization of the fuel and is 75 kcal/kg°C (iso-octane.)

Figure 2 shows the relationship between Y , the air-fuel ratio, and F , the excess air ratio, when iso-octane is used as fuel.

(b) A method to measure (x), the vaporization percentage of fuel

Automobile engines usually are equipped with intake preheaters. Therefore, it is thought that the sucked fuel totally vaporizes by the end of suction. But in those engines which are without preheaters and which use fuel of low vaporizability, such as petroleum fuel, it is inconceivable that the fuel always vaporizes completely. One method to estimate the vaporization percentage in such a case will be described here.

If a portion of the fuel is gasified by the carburetor and the mixture of air, gasified fuel and liquid fuel occupies the volume V in the cylinder; if we assume

T_a : the temperature of the air sucked in,

P : the total pressure,

P_a : the partial pressure of the air,

Fig. 2 The relationship between ΔT_{10} , a drop in the temperature due to vaporization of the fuel, F , the air excess coefficient, and Y , the air-fuel ratio

P_f : the partial pressure of the gasified fuel,
 G_f : the weight of the fuel sucked in,
 R_a : the gas constant of the air,
 R_f : the gas constant of the fuel in the gaseous state,

$$G_a = \frac{P_a V}{R_a (T_a - x \Delta T_{10})} = \frac{P}{1 + \frac{x}{Y} \frac{M_a}{M_f}} \frac{V}{R_a (T_a - x \Delta T_{10})} \quad (17)$$

If we assume that the weight of the air which occupies the same volume at the temperature T_a is G_a' the ratio of G_a to G_a' is

$$\frac{G_a}{G_a'} = \frac{1}{1 + \frac{x}{Y} \frac{M_a}{M_f}} \frac{T_a}{T_a - x \Delta T_{10}}$$

and from this, the vaporization percentage x , is

$$x = \frac{\left(\frac{T_a}{Y} \frac{M_a}{M_f} - \Delta T_{10} \right) \pm \sqrt{\left(\frac{T_a}{Y} \frac{M_a}{M_f} - \Delta T_{10} \right)^2 - 4 \frac{T_a}{Y} \frac{M_a}{M_f} T_{10} \left(\frac{G_a'}{G_a} - 1 \right)}}{2 \frac{\Delta T_{10}}{Y} \frac{M_a}{M_f}} \quad (18)$$

If M_f , the molecular mass of the fuel, is known, the value of M_a/M_f is known. Because ΔT_{10} can be found from Eq. (13) or Fig. 2, we obtain Fig. 3 by tracing the relationship between G_a/G_a' (%) and the vaporization percentage, when the air-fuel ratio is changed. If the entire fuel is gasified, the amount of air intake must increase by between 4% and 7%.

If we compare the value of G_a , in the case where only air has been sucked in by motoring of the engine immediately after firing with the value of G_a when fuel is ejected the amount of fuel which vaporises can be estimated approximately by the end of suction, and accordingly the value of ΔT_1 is found from $\Delta T_1 = x \Delta T_{10}$. In firing, high-temperature residual gas exists, and the inside wall of the cylinder is at a higher temperature than in the case of motoring. Therefore, the vaporization percentage must be higher than the estimated vaporization percentage in the case of motoring.

2.3 A rise in the temperature due to the inertia effect (ΔT_2)

In an engine where the length of the air intake pipe is zero, the inertia effect is relatively small. In a real engine, however, where the length of the carburetor is also included, there is a substantial inertia effect due to the mass of the gas in the air intake pipe. Here the process 8~I shown in Fig. 1 is supposed to handle quantitatively ΔT_2 , a rise in the temperature of mixture gas due to this inertia effect. Suppose that a cylinder whose cubic capacity is V_1 contains a gas of G_8 , P_8 and γ_8 (specific weight), and that a gas of P_i , T_i , γ_i flows into it through an intake valve and the condition of the mixture gas ultimately becomes P_1 , T_1 , γ_1 . If the weight of the gas influx is dG and its specific weight is γ , the change in the energy inside the cylinder due to the gas influx is $c_v V_1 d(\gamma T)$. This is equal to $dG c_p T_i$, the thermal energy of the gas influx. Therefore we have $dG c_p T_i = c_v V_1 d(\gamma T)$ and from $\gamma = P/RT$, we have

M_f : the molecular mass of the fuel (iso-octane=114),

M_a : the molecular mass of the air (28.96),
 and if we assume further than xG_f kg of the fuel sucked in is gasified and that $(1-x)G_f$ kg remains in liquid form, we have $P = P_a + P_f$ by Dalton's law. Now, if the volume of $(1-x)G_f$ is ignored because it is very small as compared with the total volume V , the vaporization of xG_f reduces the temperature by $x \Delta T_{10}$. Accordingly we have the following relationship regarding the air and the vaporized fuel:

$$\left. \begin{aligned} P_a V &= G_a R_a (T_a - x \Delta T_{10}) \\ P_f V &= x G_f R_f (T_a - x \Delta T_{10}) \end{aligned} \right\} \quad (15)$$

From Eq. (15) and the law of partial pressures, we have

$$\frac{P_a}{P} = \frac{P_a}{P_a + P_f} = \frac{1}{1 + \frac{x}{Y} \frac{M_a}{M_f}} \quad (16)$$

Therefore, the weight of the air which occupies the volume V , i.e., G_a , is

$$dG = \frac{1}{\kappa} \frac{V_1}{T_i R} dp \dots\dots\dots (19)$$

where c_p and c_v are, respectively, the specific heat of gas at a constant pressure and the specific heat of gas at a constant volume, and $\kappa = c_p/c_v$ is the specific heat ratio. The initial condition is G_8, P_8 and the final condition is G_1, P_1 . Accordingly, Eq. (19) is rewritten as

$$G_1 - G_8 = \frac{1}{\kappa} \frac{V_1}{T_i R} (P_1 - P_8) \dots\dots\dots (20)$$

Further, because there is almost no change in the gas constant between condition 1 and condition 8, if we assume $R = R_1 = R_8$ and if we substitute $G_1 = \frac{P_1 V_1}{RT_1}$, $G_8 = \frac{P_8 V_1}{RT_8}$ into Eq. (20), we have

$$T_8 = \frac{P_8}{P_1} \frac{T_1}{1 - \frac{1}{\kappa} \frac{T_1}{T_i} \left(1 - \frac{P_8}{P_1}\right)} \dots\dots\dots (21)$$

Because $T_1 - T_8 = \Delta T_2$, Eq. (21) can be rewritten as

$$\Delta T_2 = T_1 \left\{ \frac{1 - \frac{T_1}{\kappa T_i}}{1 - \left(\frac{P_1}{P_8}\right) - \frac{T_1}{\kappa T_i}} \right\} \dots\dots\dots (22)$$

Here the value of the specific heat ratio κ is approximately determined by the air-fuel ratio Y and the temperature at the end of suction, T_1 , as mentioned in Section 3.2. In the average engine, $Y \approx 15$ and $T_1 \approx 400^\circ\text{K}$. Accordingly, we can assume κ as 1.35 and constant. T_1 can be found from Eq. (11), and P_1 can be found from the indicator diagram. Therefore, if P_8 , the average pressure during a suction stroke, and T_i , the temperature of the inlet of the suction valve, are determined, the value of ΔT_2 is found from Eq. (22).

(a) The method to determine (P_8), the average pressure during a suction stroke

Figure 4 is the P - V diagram rewritten from the P - θ diagram which shows the pressure in the cylinder during a suction stroke. In the P - V diagram, the pressure during a suction stroke shows a relatively constant value. Because the volume V_1 of

Fig. 3 The relationship between the vaporization ratio and the mass of air intake in motoring

Eq. (19) corresponds to V_s , the cubic capacity of the effective stroke, plus V_c , the cubic capacity of the void, P_8 in Eq. (22) is assumed to be the point of crossing between the line representing the pressure in the cylinder during a suction stroke and V_s , the cubic capacity of an effective stroke. In this case, even if there is an error of 20 mmHg in the reading of P_8 , a difference of about 1% only arises because ΔT_2 in Eq. (22) is about 5°C in firing. This method of determining P_8 is sufficient for practical purposes.

(b) The method to determine (T_i), which is the temperature of the inlet of the suction valve

If P_i , which is the pressure in the suction pipe during a suction process, is assumed to be constant, T_i , which is the temperature of the inlet of the suction valve, is considered the sum of T_a , the temperature of the air sucked in ΔT_i , the drop in the temperature due to the vaporization of the fuel, and the rise in the temperature due to the transmission of heat from outside.

Practically, however, during the second half of a suction stroke, the pressure immediately before the suction valve shows a change approximately in proportion to the pressure in the cylinder because of the inertia effect of the gas in the suction pipe. The higher the speed of the engine, the greater the change of the pressure. For this reason, the temperature of the gas in the suction pipe changes slightly even if there is no supply of heat from outside.

To find ΔT_i , which is the change in the temperature which occurs when the pressure in the suction pipe changes by ΔP_i let us assume a suction system such as shown in Fig. 5. Here G_a is the weight of the air which flows into the suction pipe during a unit length of time;

G_i is the weight of the air in the suction pipe at a given moment;

G_e is the weight of the air which flows into the

The full-open speed, 2000 rpm of a 1500 cc water-cooled engine

Fig. 4 A method to find the value of P_8 from an indicator-diagram

cylinder from the suction pipe;

G_{i0} is the weight of the air in the suction pipe when $t=0$;

G_i is the weight of the air in the suction pipe.

Then we have $G_i = G_{i0} + G_a - G_e$

Its differential amount is

$$dG_i = dG_a - dG_e \dots\dots\dots(23)$$

The energy of the gas in the suction pipe at a given point of time is represented as

$$d(u_i G_i) + AP_i d(v_i G_i) = i_a dG_a - i_i dG_e \dots\dots\dots(24)$$

where u is the internal energy; i the entropy and v the specific volume.

If air is considered a perfect gas, the change before and behind the suction valve is considered an isentropic change. Therefore if $i_i = i_e$ in Eq. (24) and if du_i and dG_e are eliminated from Eq. (23) by using $u_i = i_i - AP_i v_i$, which is the basic equation of thermodynamics, we have

$$dT_i = \frac{\kappa-1}{\kappa} \frac{T_i}{P_i} dP_i - (T_i - T_a) \frac{dG_a}{G_a} \dots\dots\dots(25)$$

Near the end of a suction stroke the residual exhaust gas has totally flowed into the cylinder. Accordingly, if a case is assumed where there is no vaporization of fuel in the suction pipe and no supply of heat from outside, we have, because $T_a = T_i$,

$$\Delta T_i = \frac{\kappa-1}{\kappa} \frac{T_i}{P_i} \Delta P_i \dots\dots\dots(26)$$

where Δ is the amount of change in the temperature and pressure in Eq. (25).

The value of ΔT_2 becomes a problem when the engine is being operated with the throttle nearly full-open. As the speed of the engine rises, the minimum value of the pressure in the suction pipe during a suction stroke decreases. But in the average engine, it is about -100 mmHg.

Therefore, we know from Eq. (26) that ΔT_i is a rise of about 10°C at its maximum.

In the case of firing, there is a drop in the temperature due to the vaporization of the fuel and a rise in the temperature due to the supply of heat from outside in the form of radiation and transmission. Thus, the two counteracting factors cancel each other and it seems possible to assume approximately $T_i = T_a$. Even if T_i in Eq. (26) should change as much as 20°C , its effect on ΔT_i is 1% or less. Therefore, for practical purposes, Eq. (26) is also applicable in the case of firing.

In an engine equipped with a preheater, ΔT_3 ,

Fig. 5 Changes in the pressure in the intake pipe

which is the rise in the temperature due to preheating, is found by the method described in the next section. Its value ranges from 20 to 40°C . If ΔT_2 is found on the basis of $T_i = T_a + \Delta T_3$, we obtain a sufficiently accurate value.

Even if there is an error of 10°C in determining the value of T_i , the error which arises in the value of ΔT_2 is only about 3°C . Figure 6 shows the effect of t_i ($^\circ\text{C}$) on ΔT_2 when $T = 393^\circ\text{K}$ (120°C), $P_1 = +100$ mmHg and $P_8 = -100$ mmHg.

(c) The effects of the operating condition on ΔT_2

From Eq. (22) it is seen that the condition $\Delta T_2 = 0$ is met when $T_1 = KT_i$ and $P_1 = P_8$ (when there is no inertia effect). In firing, the relation between T_i and T_1 almost constitutes the condition which satisfies the above equation, and the value of ΔT_2 becomes small. But in motoring, there is a wide deviation from this condition. Therefore, even if the condition of the pressure during a suction stroke is the same, the value of ΔT_2 becomes considerably large. Namely, in firing, a low-temperature gas flows into the high-temperature gas in the cylinder because of the inertia effect, and lowers the temperature of the entire gas.

Therefore, the value of ΔT_2 is small. In motoring, on the other hand, there is almost no difference of temperature between the gas remaining in the cylinder and the gas which flows in by the inertia effect. Therefore, the rise in temperature is due to compression only, and the value of ΔT_2 is greater than at the time of firing.

2.4 The rise of the temperature under preheating (ΔT_3)

To find the value of ΔT_3 , the rise of the temperature due to preheating, the difference between $\Delta T_{2,3,4}$ and $\Delta T_{2,4}$ should be determined where $\Delta T_{2,3,4}$ is the value based on the mass and pressure of the air intake, measured when the engine is equipped with a preheater, and the value found from Eq. (12) and $\Delta T_{2,4}$ is found after removing, or

Fig. 6 Effects of T_i on ΔT_2

suspending the functioning of, the preheater and simulating exactly the same operating conditions, including the number of revolutions and the load, as before. Namely,

$$\Delta T_3 = \Delta T_{2,3,4} - \Delta T_{2,4} \quad (27)$$

One may think of measuring ΔT_3 directly by mounting a thermometer just before the intake valve. But with this method, the measured value is liable to be affected by the vaporization of the fuel and the radiation of heat. Therefore this method may provide some information but gives no accurate value.

2.5 The rise of the temperature due to the transmission of heat (ΔT_4)

By the methods mentioned in sections 2.3 and 2.4, the values of ΔT_2 and ΔT_3 are given. If these values are subtracted from the value of $\Delta T_{2,3,4}$ found from Eq. (12), the value of ΔT_4 , which is the rise of the temperature due to the transmission of heat from the wall of the cylinder and the top of the piston, can be found. Namely,

$$\Delta T_4 = \Delta T_{2,3,4} - (\Delta T_2 + \Delta T_3) \quad (28)$$

2.6 The rise of the temperature due to the mixing of the residual gas (ΔT_5)

As is clear from Eq. (1), ΔT_5 which is the rise of the temperature of the cold intake mixture due to mixing with the hot residual gas is

$$\Delta T_5 = T_1 - (T_a - \Delta T_1 + \Delta T_{2,3,4}) \quad (29)$$

and the value of ΔT_5 can be found by substituting the value of each member of the right side of the equation, found by the methods mentioned in the preceding sections.

3. Examinations of thermodynamic analysis

Hereinafter the writer will examine Eq. (12) given by the method of analysis introduced in this report, the various assumptions made, and the polytropic index and the results of the experimental analysis.

3.1 The hypotheses made for analysis

(a) Hypothesis 1

Hypothesis 1 assumes that there is no overlapping angle of any valve. In real engines, however, there are considerably overlapping angles of valves. But if it is assumed that despite the angle of overlapping, or within the angle as it is, the intake mixture does not escape into the exhaust pipe without staying in the cylinder or the exhaust gas which has entered the exhaust pipe does not flow back into the cylinder, the existence of the overlapping angles does not matter.

Figure 7 illustrates a case where the pressure of the residual exhaust gas is higher than the pressure in the intake pipe within the overlapping angles

of the valves. Between signs 5-6-7, expansion of the residual gas and its reflux into the intake pipe or flow into the exhaust pipe are expected to occur as follows:

The residual exhaust gas at the top dead center expands, flowing back during its expansion to point 6 where its pressure becomes equal to P_i , the pressure in the intake pipe. Later, a new suction starts at point 7 where the residual gas was totally absorbed in the cylinder.

If the pressure in the intake pipe is higher as when supercharging the engine, the residual exhaust gas is compressed till it reaches the intake pressure level. If the pressure in the exhaust pipe remains high until the exhaust valve closes as shown in Fig. 7, there may even be a reflux from the exhaust pipe. But in ordinary automobile engines, the overlapping angles of the valves are about 15 to 20°C with the top dead center as the center. Accordingly, if the ineffective lift of a valve (ineffective angle)⁽¹¹⁾ is considered, it seems necessary to consider the reflux within the overlapping angles of valves.

If we assume that-

V_0 : the capacity of a stroke at an arbitrary crank angle from the top dead center;

P_0 : the pressure in the cylinder at V_0 and

n_2 : the polytropic index of the residual gas in the first half of the suction stroke ($n_2 \approx 1.30$);

then the expansion of the residual gas due to the descent of the piston from the top dead center is presented as

$$P_0(V_c + V_0)^{n_2} = P_5 V_c^{n_2} \quad (30)$$

Therefore we have

Fig. 7 The low-pressure indicator-diagram in the case where the pressure of the residual exhaust gas is higher than the pressure in the intake pipe

$$V_{\theta} = V_c \left\{ \left(\frac{P_5}{P_{\theta}} \right)^{1/n_2} - 1 \right\} \dots \dots \dots (30)'$$

Figure 8 is a comparison of the curve shown in Eq. (30)' and the measured value when the overlapping angles of the valves are small. In this case, it seems that we have only to consider the expansion of the residual gas due to the descent of the piston. When the overlapping angles of the valves are great, the situation becomes complicated, because the pressure waves in the exhaust pipe greatly affect the inside pressure in the cylinder.

In any case, however, Hypothesis 1 may be considered valid unless there is a free passage of a new intake because the residual gas is totally absorbed in the cylinder by the end of the suction.

(b) Examinations of Hypothesis 2-4

Actually, the pressures in the intake pipe and exhaust pipe change considerably during the suction stroke and the exhaust stroke, respectively, but as shown in Section 2.3, the changes of the pressure in the intake pipe have almost no effect on the temperature of the cycle. The pressure and temperature in the exhaust pipe affect the temperature of the cycle only at the top dead center. Because these values are used in the analysis, this hypothesis is unnecessary for practical purposes, but was used for the sake of convenience when the cycle was considered in terms of an idealized model. The pressure in the cylinder can be considered nearly constant, as shown in Fig. 4. Therefore, the use of a specific heat at a constant pressure in Eq. (6) seems to be appropriate.

As is clear from Hypothesis 3 and Hypothesis 4, the stroke volume necessary for these pressure changes is about 5% of V_s , the effective stroke volume. Therefore, an error in the measurement

Full-open operation at 2900 rpm of a water-cooled automobile engine (A)
Fig. 8 Expansion of the residual exhaust gas during a suction stroke (when the overlapping angle of the value is small)

of the temperature due to these hypotheses would be relatively small.

3.2 The polytropic indices (n)

Polytropic indices n_1 and n_2 used in thermodynamic analysis are the polytropic indices between 4 and 5 on the P - V diagram and between 5 and 7 on the T - V diagram of Fig. 1. Besides, for the entire cycle, there are polytropic indices n_3 and n_4 , for the compressive stroke and the expansionary stroke respectively. The writer will attempt a brief examination of these four polytropic indices and discuss the method to determine the values of n_1 and n_2 .

Figure 9* and Fig. 10** show, respectively, the

Fig. 9 The specific heat ratio of intake mixture

Fig. 10 The specific heat ratio of combustible gas

- * In Fig. 9, the fuel was assumed to be iso-octane, and the changes of the specific heat ratio due to the changes of the air-fuel ratio have been calculated. The effects of the temperature shown by the dotted line are estimated on the assumption that they are the same as the changes of the specific heat ratio due to the changes of the atmospheric temperature, since we had no reliable data regarding their effects.
- ** Figure 10 was drawn on the assumption that the fuel is iso-octane, the specific heat ratio κ_t of the combustion gas being the excess air ratio and F the parameter and a function of the temperature. Data furnished by Awano⁽¹³⁾, M.H. Edson and C.F. Taylor⁽¹⁴⁾ were used.

values of κ_2 and κ_4 , the mixture gas and the combustion gas. In an ordinary engine, κ_4 is between the temperature at the end of suction ($T_1 \approx 400^\circ\text{K}$) and the temperature at the end of compression ($T_2 \approx 700^\circ\text{K}$). Therefore, the value of κ_3 is between 1.30 and 1.35. κ_4 is also between the maximum temperature ($T_3 \approx 2500^\circ\text{K}$) and the temperature at the end of expansion ($T_4 \approx 1000^\circ\text{K}$). Therefore, κ_4 is between 1.20 and 1.30. In a real engine, such factors as the transmission of heat between gas and the wall of the cylinder, the leak of gas from between the cylinder wall and the piston, the heat of friction and thermal dissociation must be taken into consideration. According to an analysis of an accurate indicator-diagram⁽¹²⁾, we may assume $n_3 = n_4 \approx 1.30$ (constant) regardless of the compression ratio and the load.

3.2.1 The polytropic index n_1 during an exhaust stroke

The temperature during an exhaust stroke is 400°C lower than the temperature during an expansion stroke. Therefore, the specific heat ratio κ_1 during an exhaust stroke is slightly lower than κ_4 , but the change is very small. Accordingly, it seems permissible for practical purposes to assume $n_4 = n_1 \approx 1.30$.

3.2.2 The polytropic index (n_2) during the first half of a suction stroke

The polytropic index due to the expansion of the residual exhaust gas during the first half of a suction stroke cannot be found directly from an indicator-diagram because the influx and outflow of gas at the intake and exhaust valves, respectively, and the intake and discharge of heat must be considered as mentioned above, in addition to the expansion of the residual exhaust gas due to the descent of the piston when the overlapping angles of the valves are great. But if in Eq. (12) which shows the value of $T_{2,3,4}$, the airfuel ratio is assumed to be $Y = G_a/G_f$, we have

$$G_s = G_a \frac{Y+1}{Y}$$

Therefore, in

$$G_a = \frac{Y}{Y+1} \frac{P_1 V_s}{(T_a - \Delta T_1 + \Delta T_{2,3,4})} \frac{\varepsilon}{\varepsilon-1} \times \left\{ 1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n_2} \right\} \dots\dots\dots (12)'$$

which expresses $G_s = G_a \frac{Y+1}{Y}$ as the positive function of G_a we have, when $G_a = 0$,

$$1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n_2} = 0 \dots\dots\dots (31)$$

This equation leads to

$$n_2 = \frac{(\log P_5/P_1)}{\log \varepsilon} \dots\dots\dots (31)'$$

Accordingly, the polytropic index n_2 when the residual exhaust gas is assumed as air can be obtained if P_1 and P_5 are measured from a low-pressure indicator-diagram when $G_a = 0$.*

Figure 11 shows an example of low-pressure indicator-diagrams, when $G_a = 0$. Table 1 shows the value of P_1 and P_5 which are the pressures at a given number of revolutions, with the value of n_2 calculated by substituting these values into Eq. (31)', and the value of P_1 found by substituting the measured value of P_5 into Eq. (31)' on the assumption that $n_2 = 1$ and the actually measured value of P_1 , as well as the probabilities of the errors in both.

The value of n_2 increases with the number of revolutions. This seems to be due to the fact that index n_2 approaches the index 1.4 of adiabatic change mainly because the time of suction becomes shorter while the time of suction in low-speed revolution becomes longer and the index n_2 approaches the index 1 of isothermal change. In firing, the gas used is residual exhaust gas and not air. Accordingly, the value of n_2 is slightly smaller. But at the usual speed of revolution, n_2 seems to lie between 1.2 and 1.4. Therefore, the writer used the value of 1.3 as the average value of n_2 . If P_5 is assumed as constant at 760 mmHg in order to examine the effect on $\Delta T_{2,3,4}$, the error which can arise under normal operating conditions is very small even if the value of n_2 should change by 0.1, as shown in Table 2. However, precision is lost to a considerable degree when the throttle is almost totally closed.

If the value of P_1 is found on the assumption

Table 1 The polytropic index n_1 during an exhaust stroke

rpm	P_1 mmHg	P_5 mmHg	n_2	P_1 mmHg assumed $n=1$	Measured value of P_1 Value of P_1 assumed $n=1$ error %
970	-682	0	1.24	-632	8
2 000	-676	+150	1.30	-615	10
2 300	-674	+320	1.38	-587	15

Fig. 11

* By sealing the manifold entrance of intake, motoring is performed.

$n_2=1$, as shown in Table 1, the value of P_1 is by between 50 mmHg and 80 mmHg smaller than the measured value. This means that the error is as high as 8% to 15%. From this fact it is evident that residual exhaust gas undergoes a polytropic change of $n_2 \approx 1.3$.

In Eq. (11) which presents T_1 , the value at the end of suction, the error which arises under normal operating conditions is very small even if n_2 changes by as much as 0.1, as shown in Table 3.

On the basis of the above examinations, the polytropic indices hereinafter will all be expressed as n , and n will be considered constant as: $n=n_1=n_2=n_3=n_4=1.3$.

3.3 The applicability of thermodynamic equations

The writer will apply the thermodynamic equations deduced in Chapter 2 to the temperatures in motoring in order to examine their reliability in the measurements of temperatures.

(1) The application of theoretical equations If T_1 is eliminated from Eq. (1) and Eq. (11), we have

$$\Delta T_5 = (T_a - \Delta T_1 + \Delta T_{2,3,4}) \times \left\{ \frac{\left(\frac{P_5}{P_1}\right)^{1/n} - \left(\frac{P_5}{P_4}\right)^{1/n}}{\varepsilon - \left(\frac{P_5}{P_1}\right)^{1/n}} \right\} \dots \dots \dots (32)$$

Table 2 The error in $\Delta T_{2,3,4}$ due to the change of the value of 2 by 0.1

P_1 mmHg	-600	-500	-400	-200	-100~+500
Error %	± 20	± 5	± 3	± 0.5	± 0

Table The error in T_1 due to the change of the value of n_2 by 0.1

P_1 mmHg	-500	-400	-300	-200	-100~+500
Error %	± 4	± 2	± 1	± 0.5	± 0

Fig. 12 Low-pressure indicator-diagrams for firing and motoring

But in motoring, the temperature of the residual gas is much lower than in firing. Therefore, in motoring, ΔT_5 , which is the rise of the temperature due to mixing, should be almost zero. The condition for $\Delta T_5=0$ is $T_a - \Delta T_1 + \Delta T_{2,3,4}=0$. Accordingly, it is clear from the above equation that the condition $\Delta T_5=0$ is met, when $P=P_4$.

In Eq. (6) which assumes that the heat which the intake mixture receives is equal to the heat which the residual gas loses, $T_7=T_1$ when $\Delta T_5=0$. Therefore, from Eq. (3), we have

$$T_1 = T_5 \left(\frac{P_1}{P_5} \right)^{\frac{n-1}{n}}$$

Therefore, $T_1=T_5$ when the condition $P_1=P_5$ is met. But in motoring the density of the residual air is greater than in firing, and $P_5 > P_1$. Accordingly, T_5 is slightly larger than T_1 and ΔT_5 is not zero.

Figure 12 shows low-pressure indicator-diagrams in firing and motoring at 2000 rpm. From the fact that $P_1=P_4=P_5$ in motoring, we know $\Delta T_5=0$. The fact that the results of theorization well coincide with the results of analysis by indicator-diagrams shows that our thermodynamic method of analysis introduced in this report is also applicable in the case of motoring and is correct.

(2) Comparison in regard to a rise in the temperature due to the transmission of heat

If motoring is conducted under two different conditions, namely, when the temperatures of water and oil in the engine are nearly normal and when they are high, and if ΔT_4 , a rise in the temperature due to heat transmission from the wall of each cylinder, etc., in that case, is found, the value in the first case should be lower than in the second.

Figure 13 shows low-pressure indicator-diagrams* in the case where the temperatures of both water and oil in the engine are about 80°C

Fig. 13 Low-pressure indicator-diagrams for firing and motoring with different water and oil temperature

* In this case, a special intake system without a pre-heater was used. Therefore, this diagram differs from the indicator-diagram in Fig. 12.

(hot), the case where they are both about 20°C (cold) and the case of firing, for the sake of comparison. In motoring, fuel supply was cut off in both cases in order to keep out the effect of ΔT_1 , the drop of the temperature due to the vaporization of fuel. Because not much difference is seen between the indicator-diagram for the condition "hot" and that for the condition "cold", the amount of air intake in each case is mainly affected by the different value of ΔT_4 , the rise of the temperature due to heat transmission. Figure 14 shows the measured values of P_1 , P_5 and G_a at different speeds of the engine and the method of calculating ΔT_4 and ΔT_2 , found by the aforementioned method of analysis, with the use of these values. The heating time becomes shorter with an increase in the number of revolutions. Therefore, the value of ΔT_4 becomes smaller. During the same motoring, the difference in the value of ΔT_4 becomes larger and is about 10~5°C as the speed increases if the difference in the temperature of water and oil is as much as 60°C (80°C–20°C).

Thus the value of ΔT_4 varies greatly between firing and motoring. In motoring, furthermore, the value of ΔT_4 increases as the temperatures of water and oil rise. Even under the condition "cold," the temperature of the inside wall of the cylinder rises above the temperature of water because of the friction of the piston ring, etc. Therefore, even when the temperatures of water and air intake are the same, the value of ΔT_4 is positive.

From these facts, it is evident that the method of measuring the temperature of gas based on this method of analysis is highly reliable. Besides, the author compared the value of T_1 , the temperature at the end of suction as measured by the sonic method⁽⁴⁾ practiced by Livengood and others, with the value of T_1 found by the method introduced in this report. The two values showed a high degree of correspondence. Details will be mentioned in the second report.

4. The effects of heat on the amount of air intake

The theoretical method of analysis to find the amount of air intake per cycle is the same as the method mentioned in Section 2.1.

4.1 The amount of air intake and the charging efficiency

If precise equation (10) which indicates $\Delta T_{2,3,4}$ is expressed as a positive function of G_a , the mass of air intake, on the assumption $G_s = G_a \frac{Y+1}{Y}$, we have

Fig. 14

$$G_a = \frac{Y}{Y+1} \frac{P_1 V_s}{R_1 \left\{ T_a - \Delta T_1 + \Delta T_{2,3,4} + \left(1 - \frac{c_{ps}}{c_{pr}} \right) \Delta T_5 \right\}} \times \frac{1}{\varepsilon - 1} \left\{ 1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n_2} \frac{P_1}{P_5} \right\} \dots \dots \dots (10)'$$

The member in { } in the denominator of the above equation is $\left\{ T_a - \Delta T_1 + \Delta T_{2,3,4} + \left(1 - \frac{c_{ps}}{c_{pr}} \right) \Delta T_5 \right\} = T_1 - \frac{c_{ps}}{c_{pr}} \Delta T_5$ from the definition $T_1 = T_a - \Delta T_1 + \Delta T_{2,3,4} + \Delta T_5$

As already mentioned, $c_{ps}/c_{pr} \approx 0.90$ and $R_1/R_5 \approx 0.93$ in gas near the theoretical air-fuel ratio, and the value of ΔT_5 is $30 \sim 50^\circ\text{C}$ in full-open operation and about 100°C in idling. Therefore, the value of $\left(1 - \frac{c_{ps}}{c_{pr}} \right) \Delta T_5$ is $3 \sim 10^\circ\text{C}$ and is sufficiently small compared to the value of T_1 . Therefore, if for simplicity's sake, we assume $c_{ps}/c_{pr} = 1$ and $R_1/R_5 = 1$ and $n_2 = n = 1.30$, we have

$$G_a \approx \frac{Y}{Y+1} \frac{P_1 V_s}{R_1 (T_a - \Delta T_1 + \Delta T_{2,3,4})} \frac{\varepsilon}{\varepsilon - 1} \times \left\{ 1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n} \right\} \dots \dots \dots (10)''$$

where $(T_a - \Delta T_1 + \Delta T_{2,3,4}) = T_1 - \Delta T_5$. The value of G_a found from approximate equation (10)'' is by $1 \sim 2\%$ higher than the value found from precise equation (10)'.

In view of the accuracy of the various values measured when the engine is in operation, it seems that G_a may be expressed by Eq. (10)''. As is clear from Eq. (10)'', T_5 , the temperature of residual gas has no effect on G_a . G_a is inversely proportional to $(T_a - \Delta T_1 + \Delta T_{2,3,4})$ or $(T_1 - \Delta T_5)$ but not proportional to T_1 . As stated in the foreword, the reason is that when the high-temperature residual gas mixes with the low-temperature intake mixture in the cylinder, the hightemperature residual gas shrinks as a result of cooling while the low-temperature intake mixture expands as a result of warming, but the amount of shrinkage and the amount of expansion are almost equal and accordingly they have no net effect on G_a — particularly when $c_{ps} = c_{pr}$ and $R_1 = R_5$.

The fact that G_a is inversely proportional to $(T_1 - \Delta T_5)$ is the basis for establishing a corrective equation of developed power which has theoretically a valid basis for replacing the conventional corrective equation of developed power with a dubious theoretical basis. It points out the erroneous assumption in the conventional corrective equation of developed power, that the developed power which is proportional to G_a , the mass of air intake, is inversely proportional to the temperature at the end of suction (15).

Next, it is often convenient to use the charging efficiency when one thinks of the effect of heat on the mass of air intake. We have the following equation if we express Eq. (10)'' in terms of the charging efficiency $= \eta_{ch}$

$$\eta_{ch} = \frac{Y}{Y+1} \frac{R_a}{R_1} \frac{P_1}{P_0} \frac{T_0}{(T_a - \Delta T_1 + \Delta T_{2,3,4})} \frac{\varepsilon}{\varepsilon - 1} \times \left\{ 1 - \frac{1}{\varepsilon} \left(\frac{P_5}{P_1} \right)^{1/n} \right\} \dots \dots \dots (33)$$

$$\text{where } \eta_{ch} = \frac{G_a}{G_0}, \quad G_0 = \frac{P_0 V_s}{R_a T_0}$$

G_0 is the mass of air which occupies the stroke volume V_s under normal conditions ($P_0 = 760 \text{ mmHg}$, $T_0 = 293^\circ\text{K}$ or 20°C), and R_a is the gas constant of air.

4.2 The effect of heat on the charging efficiency

4.2.1 The effect of the temperature of intake air

It is generally thought that the lower the temperature of air intake, the better the charging efficiency. The truth is that when the value of T_a decreases, the temperature difference among different parts of the engine increases and ΔT_3 and ΔT_4 are estimated to increase in general. Therefore, we must consider the overall value $(T_a - \Delta T_1 + \Delta T_{2,3,4})$. Therefore, to obtain accurate results, experimental analysis must be conducted. For example, in an engine where ΔT_3 , the rise of the temperature due to the preheating of air intake, is very high, the value of $(T_a - \Delta T_1 + \Delta T_{2,3,4})$ does not change much even if the value of T_a does. Therefore, the value of T_a scarcely affects η_{ch} .

One of the reasons why the conventional corrective equation of developed power does not coincide with the results of experiments is that the equation ignores thermal phenomena which arise in a real engine.

4.2.2 The effects of the vaporization of fuel

Generally speaking, it seems that the denser the air-fuel ratio, the higher the value of ΔT_1 , and the better the value of η_{ch} . Actually, however, the temperature of the cycle due to the air-fuel ratio changes and in addition the temperature of the wall of the cylinder, etc does. Therefore, ΔT_4 increases close to the theoretical air-fuel ratio where the average temperature of the cycle reaches its highest point. Accordingly, η_{ch} decreases to its lowest point, and should rise more or less according to whether the concentration of the intake mixture is higher or lower. Strictly speaking, however, there should be some difference between various types of engine. Therefore, the value of η_{ch} must be found by experiment in regard to each engine.

4.2.3 The effects of the rise of the temperature

($\Delta T_{2,3,4}$)

The values of ΔT_2 , the rise in the temperature due to the inertia effect, ΔT_3 , the rise of the temperature due to preheating, and ΔT_4 , the rise of the temperature due to the transmission of heat, vary with the types of engine, the operating conditions, etc. Here, an attempt will be made to find the relationship between $\Delta T_{2,3,4}$, the total sum of such thermal changes, and η_{ch} by means of Eq. (33). Figure 15 has been drawn in regard to an example where $\varepsilon=7$, $P_5=760$ mmHg abs, $Y=13.5$, $T_1=20^\circ\text{C}$, $R_1=27.8$ ($Y=13.5$, $G_5/G_8=0.06$), and $n=1.3$ and with P_1 as a parameter. In firing, the temperature of the wall of the cylinder, etc., is always higher than the temperature of air intake. Therefore, $\Delta T_{2,3,4}$ should never take a negative value. In motoring either, $\Delta T_{2,3,4}$ is never negative because the heat of friction arises on the piston ring.

The effects of ε , the compression ratio, and P_5 , the pressure of residual exhaust gas, on η_{ch} are relatively small as will be mentioned in the second report. The effects of the air-fuel ratio Y on the value of ΔT_1 are also small. Therefore, in the case where P_5 , the pressure of the intake pipe, is almost equal to P_1 , the pressure at the end of suction, as in a multicylinder engine, the value of ΔT_4 or $\Delta T_{3,4}$ is approximately known from Figure 15 if the charging efficiency of the engine is measured.

If it is assumed that $\Delta T_{2,3,4}$ is zero at $P_1=0$ mmHg, which is the case in full-open operation, η_{ch} reaches 120%. But if $\Delta T_{2,3,4}$ is considered to be 100°C in firing, η_{ch} drops to 90%. Thus, the difference amounts to 30%.

5. Conclusions

(1) The writer has classified the changes in the temperature of the gas in a cylinder during a suction stroke into five groups, namely, the change due to the vaporization of fuel (ΔT_1), the change due to the inertia effect (ΔT_2), the change due to preheating (ΔT_3), the change due to the transmission of heat (ΔT_4) and the change due to the mixing of

residual exhaust gas (ΔT_5). Then, he deduced an exact equation (10) which shows $\Delta T_{2,3,4}=(\Delta T_2+\Delta T_3+\Delta T_4)$, the rise of the temperature of the mixture gas, and an exact equation (11) which shows T_1 , the temperature at the end of suction, and their respective approximate equations (12) and (11) and he quantitatively clarified the effects of such factors as temperature, pressure and mass of the air intake on $\Delta T_{2,3,4}$ and T_1 . As a result, he could obtain the values of $\Delta T_{2,3,4}$ experimentally by substituting the measured values of the weight of the intake mixture and the pressures at various parts of the cycle into approximate equation (12) and could find the value of T_1 by substituting the values of $\Delta T_{2,3,4}$ into approximate equation (11).

(2) The writer further demonstrated a method to find ΔT_2 , the rise of the temperature due to the inertia effect, by substituting the value of T_1 , the temperature at the end of suction found from the thermodynamic analysis, and by substituting the pressure found from an indicator-diagram, etc. to find ΔT_3 , the rise of the temperature due to the preheating of air intake, as the difference between $\Delta T_{2,3,4}$ and $\Delta T_{2,4}$ through experiments performed on an engine equipped with a preheater and an engine without a preheater. Therefore, by subtracting ΔT_2 and ΔT_3 from $\Delta T_{2,3,4}$, can be found the value of ΔT_4 , the rise of the temperature due to the transmission of heat. Further ΔT_5 can be found by $T_1-(T_a-\Delta T_1+\Delta T_{2,3,4})$. Therefore, the writer demonstrated a method to separate $\Delta T_1\sim\Delta T_5$ quantitatively and find the value of ΔT_5 , where T_a represents the temperature of the air intake.

(3) To test the said thermodynamic method of analysis, the writer examined the hypotheses made for the analysis, the polytropic indices used, the applicability of Eq. (12) in the case of motoring, and the appropriateness of using the same equation in regard to the value of ΔT_4 . As a result, it could be ascertained that the thermodynamic method of analysis introduced in this report is sufficiently accurate.

(4) By deducing precise equation (10)' which presents G_a , the mass of air intake, it was learned that G_a bears no relation to the temperature of residual exhaust gas and that the effect of ΔT_5 , the rise of the temperature due to mixing of residual exhaust gas, is very small. It was clarified that approximate Eq. (10)'' which indicates G_a is adequate for practical purposes. The error in assuming that G_a is inversely proportional to T_1 , the temperature at the end of suction was pointed out. It was clarified that T_1 is inversely proportional to $(T_1-\Delta T_5)$ or $(T_a-\Delta T_1+\Delta T_{2,3,4})$. As a result, a basic concept for valid method to correct developed

Fig. 15 The relationship between $\Delta T_{2,3,4}$ and the charging efficiency

power has been proposed.

(5) The effect of ΔT_1 , ΔT_2 , ΔT_3 , and ΔT_4 on the air intake differs considerably with various types of engine and the operation conditions. When the engine is operated with the throttle fully open, we have $\Delta T_1 \approx 20^\circ\text{C}$, $\Delta T_2 \approx 5^\circ\text{C}$, and $\Delta T_3 \approx 30 \sim 50^\circ\text{C}$ but $\Delta T_4 \geq 50^\circ\text{C}$. ΔT_4 has the greatest effect on the mass of air intake.

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Discussion

T. MURAYAMA: (1) The relationship between V_5 , V_7 and V_e is not clear enough from Fig. 1. Please explain.

(2) On Line 37, page 1166, it is stated: "even if the effective compression ratio is estimated smaller than it really is, the value of P_1 in the indicator-diagram is presented larger than real". What does this mean?

(3) Line 49, page 1166, reads: " R_1 , which is the gas constant of the entire gas at the end of suction, can be calculated from the air-fuel ratio and the mixing ratio of the residual exhaust gas." Please show the way to calculate the mixing ratio.

(4) What is meant by "finding ΔT_i from the assumption $T_i \approx T_a$?"

(5) Page 1173, Section 3.3 it is stated that in motoring, ΔT_5 should be almost zero. But it is said that ΔT_5 is not zero. Isn't this a contradiction?

S. OHGASHI and Y. HAMAMOTO: (6) In Eq. (4), the number of moles changes from condition 1 to condition 4 as a result of combustion. If the effect of this change is considered, we have

$$T_1 = T_4 \frac{P_1 R_5}{P_4 R_1} \delta,$$

where δ stands for the percentage increase of the

number of moles. The value of δ varies with the air-fuel ratio and with the mass of the residual exhaust gas. If, for example, the residual exhaust gas is ignored and a theoretical air-fuel ratio is supposed, the value of δ is estimated to be about 1.05. If the air-fuel ratio is assumed as 12:1, the value of δ is about 1.1. Therefore, the value of T_1 differs by about 10% from that where δ is 1. I think this point should be considered.

(7) when the value of ΔT_3 is measured (page 1169, Section 2.4), the vaporization rate of the fuel differs according to whether or not the engine is equipped with a preheater. Therefore, strictly speaking, $(\Delta T_{2,3,4} - \Delta T_{2,4})$ does not represent ΔT_3 , does it?

T. SAITO: (8) I think that T_1 , the temperature resulting from thermal changes such as those due to the mixing with the residual exhaust gas and the transmission of heat, can be expressed as an algebraic sum of ΔT_1 , ΔT_2 , ΔT_3 , etc., only when the temperature changes in individual stages, that is, each stage is considered independent or such an order of stages is considered that the heat transmission takes place after mixing with the residual exhaust gas. What do you think of this point?

(9) During a suction stroke, the gas in the

cylinder should expand against the piston and its temperature should drop. Where in this calculation is this point considered?

(10) In the exhaust stroke 4~5, the pressure of the gas decreases because of outflow simultaneously with expansion. If this is approximated by a polytropic change, its index n_1 must be smaller than the adiabatic index (specific heat ratio). What do you think of that?

Author's closure

(1) The cubic capacity of the voids is expressed as V_v , and the volume of the residual exhaust gas at the top dead center is expressed as V_5 , $V_c = V_5$. V_7 is the volume of the residual exhaust gas at the moment when all the residual exhaust gas which once expanded to reach the intake pipe pressure has been totally absorbed into the cylinder and when the real suction of the intake mixture starts. The position of the piston at this moment is slightly lower than the upper dead center, but because the distance of lowering is considered to be very small, the relationship $V_7 = V_5 = V_c$ is assumed in Fig. 1 which expresses the model of a cycle.

(2) In Fig. 4, if for some reason, ϵ , the effective compression ratio, is estimated to be small, the values of V_s and ϵ become small, but the value of P_1 at this volume becomes large. This fact shows that the functional relationship between P_1 and ϵ and V_s in Eq. (12) which expresses $\Delta T_{2,3,4}$ has almost no effect on the value of $\Delta T_{2,3,4}$.

(3) R , the gas constant of the intake mixture, can be calculated as the function of R_f , the gas constant of the fuel vapor, R_a , the gas constant of air, and Y , the air-fuel ratio, or F , the excess air ratio from the law of mixture gas. R_5 , the gas constant of the combustible gas, can be calculated from the temperature and the chemical equilibrium constant of the combustible gas by assuming the composition of the gas produced by combustion as CO_2 , CO , H_2O , H_2 , and A .

The Append.-Fig. 1 indicates the values of F or Y and R^* and R_5 in the case where the fuel is iso-octane.

R_1 , the gas constant of the entire gas at the end of suction, is expressed by the following equation as a function of R^* , R_5 , G_s which is the mass of the intake mixture and G_5 which is the mass of the residual exhaust gas:

$$R_1 = \frac{G_s}{G_s + G_5} R^* + \frac{G_5}{G_s + G_5} R_5 = \frac{R^* + \frac{G_5}{G_s} R_5}{1 + \frac{G_5}{G_s}} \quad \dots\dots(i)$$

Then, G_5/G_s can be found as a function of P_4

and P_5 from

$$\frac{G_5}{G_s} \left\{ \frac{1}{1 - \frac{1}{\epsilon} \left(\frac{P_5}{P_4} \right)^{1/n} - 1} \right\} \dots\dots(7)'$$

Therefore, the value of R_1 is obtained by finding the values of R^* and R_5 at the air-fuel ratio at the time of the test and substituting them into Eq. (i). The purpose of the Append.-Fig. 2 is to find the relationship between G_5/G_s and R_1 with the excess air ratio F as a parameter.

(4) T_i , the temperature of the gas in the intake pipe during firing, does not change if the drop in the temperature due to the vaporization of fuel and the rise in the temperature due to transmission of heat from outside cancel each other. In that event, $T_i = T_a$. Accordingly, ΔT_i may be found from Eq. (26) assuming that there is no net effect of vaporization and transmission.

(5) Page 1173, Section 3.3, the writer meant to say that the value of ΔT_5 in motoring is very

Append.-Fig. 1 F , the excess air ratio coefficient; MR , the air-fuel ratio; R^* and R_5

Append.-Fig. 2 The relationship between R_1 and G_5/G_s with F as a parameter

small and nearly zero when compared with the value of ΔT_5 in firing. It is stated that the value of ΔT_3 is not zero even in motoring, strictly speaking.

(6) Under condition 1 and condition 4, respectively, the following equations apply:

$$P_1 V_1 = (G_s + G_b) R_1 T_1 \dots \dots \dots (ii)$$

$$P_4 V_4 = (G_s + G_b) R_5 T_5 \dots \dots \dots (iii)$$

where R_1 stands for the gas constant of the entire gas at the end of suction, and R_5 stands for the gas constant of the combustible gas.

Because $V_1 = V_4$, we have, from Eq. (ii) and Eq. (iii),

$$T_1 = T_4 \frac{P_1}{P_4} \frac{R_5}{R_1} \dots \dots \dots (iv)$$

since the gas constant is expressed as $R = \frac{848}{M}$,

where M is the molecular mass of the gas. Therefore, R_5/R_1 represents the ratio of the molecular mass of the entire gas to that of the combustible gas, i.e., the ratio of the number of moles. As already pointed out in answer to Mr. Murayama's question, gas constants in which the changes of the number of moles are taken into consideration are used for R_1 and R_5 . Accordingly, no error can result from the changes of the number of moles.

(7) In a four-stroke gasoline engine, even without a preheater, the fuel is believed to vaporize totally by the end of suction because the polytropic index during a compression stroke is fixed and also because the analysis of the suction stroke indicates that^(a).

$\Delta T_{2,3,4}$ and $\Delta T_{2,4}$ in the report represent the rises of the temperature due to the transmission of heat by the end of suction. Therefore, the value of ΔT_3 can be found from the equation $\Delta T_3 = \Delta T_{2,3,4} - \Delta T_{2,4}$ regardless of the difference of the vaporization rate of the fuel in the intake pipe and the cylinder due to the existence or nonexistence of the preheater.

(8) The amount of air intake depends on the pressure and temperature at the end of suction and the amount of the residual exhaust gas. Therefore, as far as the effects of the temperature are concerned, neither the process of mixing the new air intake and the residual exhaust gas nor the process of the transmission of heat between the gas in the cylinder and the wall of the cylinder poses any problem if T_1 , the temperature at the end of

suction, is correctly determined. Namely, during a suction stroke, cold intake mixture is absorbed into the cylinder containing hot residual exhaust gas. Therefore, heat escapes into the wall of the cylinder from the area containing a lump of hot residual exhaust gas, while heat enters the area containing a lump of cold intake mixture.

The question is how much net outflow or influx of heat does take place. It does not matter where the outflow or influx of heat occurs.

In this report, it is assumed that the outflow and influx of heat always occur in a new intake mixture and that there is no influx or outflow between the residual exhaust gas and the wall of the cylinder. Therefore, $\Delta T_{2,3,4}$ is accurate, but ΔT_5 is the net result of the thermal change due to the transmission of heat out of the residual exhaust gas.

The analysis was conducted by taking into consideration the net transmission of heat between the intake mixture and the residual exhaust gas. Therefore, T_1 , which was found as an algebraic sum of ΔT_1 , ΔT_2 , ΔT_3 , etc., is correct.

(9) The changes of the gas passing through the intake pipe are considered isoenthalpic changes, and the work of the piston during a suction stroke is considered to be consumed in the decline of the pressure of the gas when the gas passes through the intake pipe. Therefore, the work of the piston is considered to be unrelated to the temperature of the gas.

(10) The temperature of gas during the exhaust stroke 4~5 is lower than that during an expansion stroke. Therefore, 1, which is the specific heat ratio during the exhaust stroke 4~5, is slightly smaller than that during an expansion stroke. (See Fig. 10) Furthermore, the polytropic index n_1 should be larger than 1 as a result of heat loss. But it is difficult precisely to determine the value of n_1 .

When one thinks of the average temperature of the gas in the exhaust pipe, one may have to assume $n_1 < \kappa_1$ because the amount of outflow and the temperature of gas within each crank angle change. But to determine the temperature of the gas in the cylinder, one has only to consider the specific volume of the gas in the cylinder regardless of whether or not there is a gas outflow. Therefore, such a consideration seems to be unnecessary.

Even if there is some error in determining n_1 , the effects of n_1 on $\Delta T_{2,3,4}$ and T_1 are very small, as pointed out in the report.

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