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Gas Exchange Process of Four-Stroke Spark Ignition Engines
under the Condition of Partial Load, at Medium or
Low Speed (2nd Report) *

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It is a well known fact that the heat transfer from each part of the engine influences the volumetric efficiency. There have been already shown many papers treating the problems of the heat transfer at the compression and expansion strokes, but few works regarding this problem have been made at the intake process.

In this paper, the averaged formulas of heat transfer coefficient at the intake process are obtained with reference to the past studies at the compression and expansion stroke, and the prediction method of the volumetric efficiency including the heat transfer effect is shown.

Good agreement between the predictions and experiments on the intake charge shows that this prediction method is appropriate to obtain the volumetric efficiency with heat transfer effect.

1. Introduction

In the previous paper part 1⁽¹⁾, the inlet cavity model and the method of calculation for it were shown. And it was confirmed that the intake air charge could be estimated almost qualitatively under the condition of partial load, low or medium speed at motoring conditions.

On the other hand it is well known that at firing condition, the wall temperature rise of the engine caused by the heat transfer exerts an influence on the intake charge.

Many studies have been made at the combustion process as the aim of the combustion analysis or the thermal load problems, but few works have been made at the gas exchange process.

In this paper the averaged heat transfer equations for the intake process are obtained with reference to the previous studies on heat transfer at the combustion and expansion process.

The effect of heat transfer on the intake charge is investigated and the method of predicting the intake charge taking the heat transfer into account is examined.

2. Nomenclature

A : Heat equivalent of work
 A : Area (with subscript)
 C : Constant
 c_m : Mean piston speed
 c_n : Specific heat of polytropic change

c_v : Specific heat at constant volume
 D_H : Equivalent diameter
 D_v : Diameter of valve seat
 d : Representative diameter
 F : Area
 F_n : Flow factor
 G : Weight
 g : Acceleration of gravity
 h : Heat transfer coefficient
 k : Constant
 m : Exponent index
 N_u : Nusselt number
 n : Polytropic index
 P : Pressure
 Q : Heat flow
 R : Gas constant
 R_e : Reynolds number
 T : Temperature
 t : Time
 u : Flow velocity
 V : Volume
 κ : Ratio of specific heat
 ν : Viscosity
 λ : Thermal conductivity
 θ : Crank angle

Subscripts

a : Atmosphere
 c : Cylinder
 e : Exhaust
 g : Working fluid
 i : Intake
 t : Throttle
 v : Valve
 w : Wall surface

3. Equations used in calculation

The assumptions for calculation are nearly the same as in our previous paper part 1. The difference is that the present calculations take heat transfer from each wall surface into consideration.

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Furthermore, it is assumed that at firing condition the inlet cavity model will hold for the exhaust system as it does for the intake system(2).

Figure 1 shows a schematic diagram of the intake and exhaust system.

3.1 Energy equation of cylinder, intake and exhaust manifold

From the law of conservation of energy the following equation representing the equation covering the cylinder is determined. The flow direction at the valve causes changes in the temperature and subscript in the first term on the righthand side of the equation, but as they are the same as in the previous paper they are omitted.

$$d(P_c V_c) + (\kappa - 1) P_c dV_c = \kappa R (T_c dG_{ci} - T_c dG_{ce}) + \frac{\kappa - 1}{A} (dQ_{ci} + dQ_{ce}) \quad (1)$$

For the inlet manifold

$$V_i dP_i = \kappa R (T_i dG_{ai} - T_i dG_{ie}) + \frac{\kappa - 1}{A} dQ_i \quad (2)$$

For the exhaust manifold

$$V_e dP_e = \kappa R (T_e dG_{ei} - T_e dG_{ea}) + \frac{\kappa - 1}{A} (dQ_{ei} + dQ_{ea}) \quad (3)$$

In the related equations the direction of the flow at the valve determines which equation must include the heat flow Q_{vi} and Q_{ve} at the valve in the heat transfer term. That is to say for the regular flow at the intake valve, heat flow is transmitted to the gas flowing into the cylinder from the inlet manifold. Therefore Q_{vi} is added to the righthand of Eq. (1), but in the reverse flow that should be added to the righthand of Eq. (2). In the same way for the heat flow at the exhaust valve, Q_{ve} is added to the righthand of Eq. (3) for the regular flow from the cylinder to the exhaust manifold, but for the reverse flow it should be included in the righthand side of Eq. (1).

3.2 Mass flow rate through throttle, inlet and exhaust valves

Since the engine operating condition causes hardly any changes in the temperature of the throttle valve, which remains near the atmosphere temperature, the flow is assumed to be adiabatic. The state before the throttle is represented by subscript 1 and the state after it by subscript 2, and the mass flow rate is given by the following

$$dG_i = CA_i \sqrt{\frac{2g}{RT_i}} \times P_i \sqrt{\frac{\kappa}{\kappa - 1} \left[\left(\frac{P_2}{P_i} \right)^{\frac{2}{\kappa}} - \left(\frac{P_2}{P_i} \right)^{\frac{\kappa + 1}{\kappa}} \right]} dt \quad (4)$$

On the other hand at firing condition the temperature of the inlet and exhaust valve becomes considerably high, so it would be unreasonable to assume the flow to be adiabatic. If this flow is

assumed to make a polytropic change the mass flow rate can be expressed by the following equation.

$$dG_e = CA_e \sqrt{\frac{2g}{RT_e}} \times P_e \sqrt{\frac{n}{n - 1} \left[\left(\frac{P_2}{P_e} \right)^{\frac{2}{n}} - \left(\frac{P_2}{P_e} \right)^{\frac{n + 1}{n}} \right]} dt \quad (5)$$

where $n = 1$

3.3 Heat transfer equation

There are many studies on heat transfer in the engine cylinder during the compression and expansion strokes, and similar definitions of the heat transfer coefficient to those in the case of a steady-state heat transfer in a forced convection have been made. If it is assumed that a similar equation regarding the gas exchange process holds true, the heat flow is represented by the following equations.

$$dQ_c = A_c h_c (T_{cw} - T_{cg}) dt \quad (6)$$

$$N_u = \frac{h_c d_c}{\lambda} = CR_u \quad (7)$$

It is assumed that for the inlet manifold and inlet valve a steady-state heat transfer equation can be applied in the same way as Eqs. (6) and (7).

4. Heat transfer equations employed for prediction

4.1 Heat transfer in the inlet manifold

Many studies have been made on heat transfer at the steady-state forced convection in a tube and the heat transfer coefficients have also been shown.

The correction terms of the coefficient for the pipe diameter and length ratio⁽³⁾ were added to these results with the Prandtl number assumed to be constant to form the following standard equations.

$$N_{ui} = 0.025 R_{ui}^{0.8} \quad (8)$$

$$R_{ui} = \frac{U_i d_i}{\nu} \quad (9)$$

Here the representative length d_i is taken to be the pipe diameter while the representative speed is the mean flow velocity of the inlet manifold obtained from the mean mass flow rate at the throttle valve and inlet valve.

With regard to the constants in Eq. (8) the steady-state correction coefficient

Fig.1 Model of inlet and exhaust systems

ents have been shown for the adjustments at the vicinity of the entrance (4-6) and there is a possibility of that changing in the range of 0.020 to 0.035.

4.2 Heat transfer at valve

Several studies have been made on the steady-state heat transfer at the inlet valve. A comparison of these with the authors' experiments was made and the relationships between the Reynolds number and the Nusselt number are shown in Fig. 2.

The line (a) is the empirical expression that Kapadia et al. (7) obtained. The line (b) is that of Engh (8), and (c) is obtained by the authors. The lines (a) (c) and (b) differ originally in the definition of the Reynolds number, but in the diagram adjustments were made before making comparison. The value of (b) was considerably large compared to (a) (c) and furthermore the tendency with the valve lift was in reverse. The exponents of the Reynolds number in (a) and (c) were nearly equal, and the tendencies by the valve lift corresponded with each other. Here the empirical expression by the authors whose area of heat transfer was known was employed as the standard equation.

$$Nu_v = 0.68 \times 10^{-3} Re_v^{1.16} \quad (10)$$

$$Re_v = \frac{u_v D_H}{\nu} \quad (11)$$

$$D_H = \left[1 - \left\{ 1 - \frac{A_v}{(\pi/4) D_i^2} \right\}^{1/2} \right] D_i \quad (12)$$

4.3 Heat transfer in the cylinder

Almost all the studies on heat transfer in the engine cylinder have been made at the compression or expansion stroke. There are some problems in employing them unchanged in the gas exchange process, but they are of value as reference in estimating the general tendency of the heat transfer effect.

In the gas exchange process the changes in the piston displacement and the changes in the intake charge due to the valve lift will cause the changes in every instant in the flow velocity in the cylinder that exerts a great influence on heat

Fig.2 Heat transfer coefficient at inlet valve

transfer. Moreover there are many variations in wall surface temperature both in location and in time, so a formula that gives the instantaneous heat flow cannot be required.

In this paper in a similar method to that in reference (9), the mean heat transfer coefficient during the intake process was assumed and the heat flow Q was assumed to be given by the following.

$$Q = \bar{h}_c \left[\int_{\theta_0}^{\theta_c} \int_{F_0}^{F_c} T_w(\theta) dF d\theta - \int_{\theta_0}^{\theta_c} T_s(\theta) F(\theta) d\theta \right] \quad (13)$$

$$= \bar{h}_c \int_{\theta_0}^{\theta_c} \{ T_w(\theta, z) - T_s(\theta) \} F(\theta) d\theta \quad (14)$$

The mean heat transfer coefficient $\bar{h}_c$ during the gas exchange process was estimated taking the value of the compression or expansion stroke as reference.

It is generally assumed that at the compression or expansion stroke the heat transfer coefficient in the convective heat transfer is given by Eq. (7). The exponent m of the Reynolds number is in the range of 0.7 to 0.8, and for the constant c depending on the investigators the values have been suggested in the range of 0.035 to 0.26 (10-15). These differences seem to be caused by the shape of the combustion chamber and other engine specifications, but another large factor is the definition of the Reynolds number.

In these studies there are many cases that the mean piston speed C_m is defined

Fig.3 Flow velocity in the cylinder at intake process

Fig.4 Mean flow velocity in the cylinder

as the representative speed in determining the Reynolds number but there are some cases that C_m is multiplied by a constant so that the flow velocity in the cylinder U is defined as $U = k \cdot C_m$. When these definitions of the Reynolds number were converted to be the same, the constant c fell within a much narrower range of 0.1 to 0.3.

Several measured values of the flow velocity in the cylinder during the intake process have been shown in references (16) to (19).

In the same way as these results the experimental values of the flow velocity by the authors at a position 8 mm below the cylinder head and 0.75R from the center are as shown in Fig. 3. The mean flow velocity at this point during the intake stroke is roughly proportional to the mean piston speed as shown in Fig. 4. The velocity at this measuring point cannot be said to give the representative speed in the cylinder, but it may be assumed that it shows the tendency of the mean flow velocity. Therefore, the mean flow velocity during the intake process except on very low speed can be represented by multiplying the mean piston speed by a constant, and it may be expressed as $U = k \cdot C_m$.

From the above consideration, the following average equation of the compression and expansion period hitherto employed is used as the standard equation of the heat transfer coefficient.

$$N_s = 0.2 R_s^{0.8} \quad (15)$$

$$R_s = \frac{u d}{\nu}, \quad u = k C_m \quad (16)$$

It seems that k , the proportional constant, will change over a wide range depending on the kind of diesel or gasoline engine, combustion method, shape of the port, and use of a shrouded valve and so on.

That the mean flow velocity at the intake process (18) was twice as large as C_m was found from measuring the swirl of the steady-state flow by the cylinder head of the engine used in this experiment. The data in reference (19) also showed that the flow velocity in the direction of the circumference at the intake process was about twice the velocity during the expansion process, therefore, the standard value of k was taken to be 2.

The predictions were made on the intake charge with heat transfer equations,

Fig.5 Schematic diagram of experimental apparatus

and comparison with the experimental values was made in order to examine the heat transfer coefficient and the prediction of the intake charge.

5. Experimental apparatus, method and results

A schematic diagram of the experimental apparatus is shown in Fig. 5 and the specifications of the tested engine and experimental conditions are shown in Table 1.

A single cylinder diesel engine was used in the experiment and a circular orifice was placed at the position of the throttle valve. The ratio of the cross section area of the orifice to that of the inlet manifold which was equivalent to the throttle valve opening was taken to be the throttle ratio, and several sizes of the orifice were used.

The reason why a diesel engine was employed here was that the load could be changed without relating to the throttle degree and the intake charge, and furthermore there was no effect of the vaporization of fuel during the intake stroke. Thus the heat control and analysis of the engine were easy and there was no difficulty in changing the wall temperature conditions.

The intake charge was measured with a circular orifice installed before a surge tank of about 500%, and the cylinder pressure, inlet and exhaust manifold were measured with a low pressure indicator (20). The wall temperatures at various

Table 1 Engine specifications and experimental conditions

ENGINE SPECIFICATIONS	
SINGLE CYL. FOUR STROKE C. I. ENGINE	
CYL. BORE x STROKE	96 x 113 mm
STROKE VOLUME	818 cc
COMPRESSION RATIO	19
VALVE TIMING	IN. OPEN 17° BTDC
	CLOSE 43° ABDC
	EX. OPEN 50° BBDC
	CLOSE 10° ATDC
OPERATING CONDITIONS	
ENGINE SPEED	500 - 2000 RPM
THROTTLE	0.05, 0.1, 0.2, 0.3
COOLANT TEMP	20, 50, 80 °C
LOAD	LOW, MID., HIGH

Fig.6 Position of wall temperature measurement

points of the engine were measured with the thermocouples as shown in Fig. 6. To reduce the pulsating effect in the exhaust system at firing condition, a 67ℓ surge tank was attached to the exhaust system.

By changing the cooling water temperature in motoring and by changing the injection fuel amount and cooling water temperature in firing operation, the wall temperature condition was changed over a wide range, and the intake charge and pressure at various throttle degrees and engine speeds were measured.

In the case of motoring the temperature difference, which was created by the coolant temperature, between the wall temperature and working fluid and changes in engine speed caused changes in the heat transfer coefficient and changes in heat flow. In firing operation by installing a surge tank at the exhaust system the effects of the exhaust pulsation on the volumetric efficiency were hardly felt, then the change of the load might have hardly any influences except for changing the wall temperature. One example of the experimental results is shown in Fig. 7.

6. Examination of heat transfer equations

A comparison between the predicted values and experimental ones of the volumetric efficiency (η_v) obtained from the energy equations with heat transfer and examinations of the heat transfer equations and the influences from the wall temperature in heat transfer were made. In these calculations the standard condition was a throttle ratio 0.1, and engine speed 1000 rpm, while the wall temperature unless indicated specially was based on the average value at firing conditions.

6.1 Heat transfer in inlet and exhaust manifold

In the equation of the heat transfer coefficient (8) it was shown that the constants might be changed within a certain limit. This limit is the value in

Fig.7 An example of experimented volumetric efficiency

case of steady-state flow but the influence of such changes in the volumetric efficiency is about 0.18% so it seems that Eqs. (8), (9) may be taken to represent the heat transfer coefficient in the inlet manifold.

Regarding the heat transfer coefficient at the inlet valve, the empirical expression of the authors and that of reference (7) were employed to determine the influence on the volumetric efficiency. But the difference was only about 0.17% and was not so large. Here Eqs. (10) to (12) where the heat transfer area was known were used.

6.2 Heat transfer in the cylinder

When the measured values and predicted ones were compared at the standard condition, the predicted values were somewhat larger so the constant in Eq. (15) was changed for 0.2 to 0.25 and the calculations were made again. Figure 8 shows a comparison between the experimental values (○), and predicted ones (●) when the wall temperature condition was changed. The trend of the volumetric efficiency by the heat transfer in both cases can be said to correspond approximately. Table 2 shows the abbreviations of the experimental conditions.

As previously noted the value of k in Eq. (16) may change to a considerable extent depending on the engines. The influence of k is equivalent to changing the constant in Eq. (15). Table 3 shows the effect on the volumetric efficiency when k was changed from 1.0 to 3.0. This shows that the determination of the representative speed in the cylinder has a considerably large effect on the volumetric efficiency. Therefore, when the tested engine differs, especially depending on the flow velocity in the cylinder, Eqs. (15)

Table 2 Experimental conditions and their codes

CONDITION	COOLANT TEMP.	LOAD	CODES
MOTORING	20° C	—	M L
	50° C	—	M M
	80° C	—	M H
FIRING	20° C	LOW	F L L
		HIGH	F L H
	80° C	LOW	F H L
		HIGH	F H H

Fig.8 Comparison between experiments and predictions in η_v

(16) cannot be said to represent a general value. According to reference (13), the effect of the average flow velocity in the cylinder on the Nusselt number is said to be given by $Nu_0(1 + 0.062 U_m/Cm)$ in contrast with the case of Nu_0 when there is no swirl.

The influence of heat transfer from all parts of the engine on the intake charge was calculated from these standard equations, and the results are as in Table 4 which confirmed that the heat transfer in the cylinder had a great influence on the volumetric efficiency as hitherto had been reported.

6.3 Difference in heat transfer coefficient by location in the cylinder

The trends of the predicted volumetric efficiency and experimental ones shown in Fig. 8 seemed to show a good correspondence. But as examined in detail, the trends of the volumetric efficiency in high temperature coolant at motoring (MH), and in low temperature coolant at firing (FLL) turned out to be reversed. As described before this was not caused by the inertia or pulsation effect of the exhaust system⁽²⁾, but mainly caused by the change in the wall temperature conditions.

When the wall temperatures of the cylinder liner, piston and cylinder head were compared at the motoring and firing conditions, the temperature of the piston at firing was considerably higher than at motoring and that of the head was roughly equal, while that of the liner was, on the other hand, lower. The difference in the volumetric efficiency has its cause either in the error of the wall temperature measurements or in the difference of the heat transfer coefficient at the piston, liner, and head.

No extreme difference could be detected in comparisons with the measured values of the temperature in other experimental examples⁽²¹⁻²⁴⁾, and it was difficult to assume that there was a great error in the representative temperature. On the other hand in a comparison of the heat flux in the head and liner during the

expansion stroke, the latter is said to be about 1/4 of the former in reference (4). According to the data of the heat flux introduced in this, the both values of the heat flux during the intake stroke are small and uncertain, but they show roughly the same values. If the two values are equal, this would mean that in the intake stroke the heat transfer coefficient of the liner would be 1.5 to 2.0 times larger than the others. This assumption seems to have the tendency to reduce the difference between the experiments and predictions. Therefore, calculations were made in the case of piston and head from Eq. (15), but for the liner the heat transfer coefficient was doubled to form Eq. (15)'.

$$Nu = 0.4 Re^{0.8} \dots \dots \dots (15)'$$

The results are shown in Fig. 8 (o).

The experiments and predictions showed a better correspondence when the heat transfer coefficient was divided into two equations (15) and (15)'. Similar results were obtained in the different conditions of other throttle ratios.

Thus in this engine, the more precise prediction values of the volumetric efficiency can be obtained by the divided heat transfer coefficient.

The heat transfer from the cylinder liner is one of the important factors in the cylinder, and there is a larger difference in temperature between the upper

Fig.9 Polytropic index changed by valve lift

Fig.10 Flow factor changed by n

Table 3 Effect of the constant k

COEFFICIENT k	$Nu/Nu_{k=2}$	η_v %	$\eta_v - \eta_{v_{k=2}}$ %
1.0	0.57	81.63	3.96
1.5	0.79	79.38	1.71
2.0	1.00	77.67	0
2.5	1.20	76.37	-1.30
3.0	1.38	75.35	-2.32

THROTTLE = 0.1 N = 1000 rpm

Table 4 Effect of heat transfer from each part on η_v

Heat Transfer Conditions	η_v %	$\eta_v - \eta_{v_{std}}$ %
With All ΔQ	79.35	0
Without ΔQ from In. Manifold	79.60	0.25
Without ΔQ from In. Valve	79.65	0.35
Without ΔQ from Cylinder	89.57	10.22

THROTTLE = 0.1 N = 1000 rpm

and lower parts of the liner, therefore, it would be controversial to represent the liner temperature by the measured value at one point, or by a simple average of the upper and lower parts.

The temperature distribution of the piston and cylinder head should be considered but the calculations became more complicated and in considering heat transfer in the cylinder, the temperature distribution was not so wide as compared to the liner so it was assumed to be constant. The changes of the wall temperature during the intake process can be also neglected.

6.4 Polytropic index at the inlet valve

It is known that heat transfer at the valve causes a decrease in the mass flow rate⁽⁷⁾ (25), and it was assumed that the gas passing through the valve made a polytropic change since the temperature of the inlet valve became so high at firing condition. From the state before and after the valve, the heat flow Q can be expressed as

$$\Delta Q = c_n(T_2 - T_1)\Delta G \\ = \frac{n-k}{n-1} c_n T_1 \left\{ \left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right\} \Delta G \dots (17)$$

On the other hand the heat flow and mass flow rate can be obtained from Eqs. (6), (10) to (12). By these simultaneous equations the polytropic index n can be determined. The obtained value n by this method is represented as a function of the valve lift and pressure ratio, and the relations are as shown in Fig. 9.

This figure shows that n makes a considerably large change depending on the condition, and the influence of n on the mass flow rate can be investigated by Fig. 10, which shows the change of the flow factor F_n . Generally when the throttle valve is throttled to a considerable extent there is hardly any pressure difference before and after the inlet valve during the intake stroke and the pressure

Table 5 Effect of the polytropic index on η_v

POLYTROPIC INDEX	n_1	1.4	1.3	variable	variable
VOLUMETRIC EFF.	η_v	0.7752	0.7752	0.7777	0.7778
CYL. PRESSURE	P_c	Error 0.001	kg/cm ² at Max.		

$V_c = 818 \text{ cc}$ $N = 1000 \text{ rpm}$
THROTTLE = 0.1

Table 6 Effect of wall temperature on η_v

THROTTLE = 0.1 $N = 1000 \text{ rpm}$					
PART	STD WALL TEMP °C	WALL TEMP °C	η_v %	η_v %	$\eta_v - \eta_{std}$ %
IN. MANIFOLD	60	40	79.44	+	0.09
		80	79.24	-	0.11
IN. VALVE	190	140	79.43	+	0.08
		240	79.28	-	0.07
CYL. HEAD	140	100	79.71	+	0.36
		180	79.00	-	0.35
PISTON	180	150	79.62	+	0.27
		210	79.08	-	0.27
CYL. LINER	100	80	81.15	+	1.80
		120	77.53	-	1.82

ratio is nearly equal to 1. Therefore as Fig. 10 shows, there are hardly any effects on the mass flow rate. In the cases of the pressure ratio being 0.99 and $n = 1.4$ (adiabatic) or $n = 0.2$, the difference in the mass flow rate is only about 3%.

Table 5 shows the intake charge being calculated when the polytropic index n at the inlet valve is constant and when n to be changed by the pressure ratio and the valve lift. It shows that even if n is assumed to be constant, this will hardly have any effects on the volumetric efficiency. Therefore, it is almost meaningless to determine n in each condition and at each step of the calculations. To shorten the time required for the calculations, it would be appropriate to assume n to be constant. The index $n = 1.3$ is employed here to distinguish from the adiabatic change.

6.5 Effect of wall temperature on volumetric efficiency

Table 6 shows the effect of the changes in the wall temperature at various points on the volumetric efficiency. The width of the temperature changes is that of the experimental values at firing conditions. These show that the heat transfer, especially from the cylinder liner, has a great effect on volumetric efficiency.

Fig.11 Comparison of volumetric efficiency (motoring, throttle 0.1)

Fig.12 Comparison of volumetric efficiency (firing, throttle 0.1)

7. Comparisons between experiments and predictions

Figs. 11 to 13 show the examples of comparisons between the experimental values and predicted ones.

Fig. 11 shows the case when the cylinder wall temperature was changed by the coolant temperature under the motoring condition at a throttle ratio of 0.1.

Fig. 12 shows the case that the wall temperature was changed by the load and coolant temperature at the firing conditions. Figure 13 shows the similar comparisons when the throttle ratio was changed. The drop in volumetric efficiency due to the wall temperature rise is generally larger at lower speeds. The rate of the drop is also larger at low speeds.

In these comparisons, the trends of the experiments and predictions show good agreement except for the throttle ratio of 0.3 at 2000 rpm. It appears that the prediction method described here can predict the influence of heat transfer on the intake charge.

8. Conclusions

The prediction method of the volumetric efficiency taking the averaged heat flow, that has a great influence on the intake charge, into consideration at partial load and at low or medium speed was described.

An inlet cavity model was used for the inlet system and the averaged heat transfer coefficient referred to the past studies at the compression or expansion stroke was used to estimate the volumetric efficiency and heat flow.

Through precise examinations it was confirmed that by employing the method in this paper, qualitative predictions considering the influence of heat transfer on the intake charge could be made and the validity of this prediction was clarified.

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Fig.13 Comparison of volumetric efficiency
(firing, throttle 0.2)

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References

- (1) Tasaka, H. and Matsuoka, S., Trans. Japan Soc. Mech. Engrs. (in Japanese), Vol.40, No.335(1974-7), p.2034
- (2) Tasaka, H. and Matsuoka, S., Preprint of Japan Soc. Mech. Engrs. (in Japanese), (Yamanashi Pref.), (1974-11), p315
- (3) Hause, H., Z.VDI, Vol.86(1942), p.353
- (4) McAdams, W.H., Heat Transmissions, (1954), McGraw-Hill
- (5) Chollete, A., Chem. Engng., Prog., Vol.44(1948), p.81
- (6) Boelter, L.M.K. et al., NACA TN, 1451(1948)
- (7) Kapadia, D. N. and Borman, G. L., SAE paper, 670479(1967)
- (8) Engh, G. T. and Chiang, C., SAE paper, 700501(1970)
- (9) Brosinsky, H., VDI Forsch., Heft 440 (1953)
- (10) Annand, W. J. K., Proc. Inst. Mech. Engr., Vol.177, No.36(1963)
- (11) Woschni, G., MTZ Heft 26, Nr 4, (1965)
- (12) Woschni, G., SAE paper, 670931(1967)
- (13) Zapf, H., MTZ Heft 30, Nr 12, (1969)
- (14) Le Feuvre, T. et al., SAE paper 690464, (1969)
- (15) Oguri, T. and Aisawa, Preprint of Japan Soc. Mech. Engrs., (in Japanese), No.730-17, (1973), p.201
- (16) Seminov, E. S., Combustion in Turbulent Flow, (1963), Isr.Prog.Sci.Tr.
- (17) Hamamoto, et al., Preprint of Japan Soc., Mech. Engrs., (in Japanese), No.700-16(1970), p.83
- (18) Ohigashi, et al., Preprint of Japan Soc. Mech. Engrs., (in Japanese), No.720-14(1972), p.49
- (19) Asanuma, T. and Obokata, Preprint of JARI 7th Res. Meeting, (in Japanese), (1972-10), p.45
- (20) Matsuoka, S. and Tasaka, H., Jour. Japan Soc. Mech. Engrs. (in Japanese), Vol.72, No.601, (1969), p.204
- (21) Pflaum, W., MTZ Heft 22, Nr 3, (1961)
- (22) Yoshida, S., J. of Internal Combustion Engine, (in Japanese), Vol.2, No.7, (1963)
- (23) French, C.C.J., PIME Symp. on Thermal Loading of D.Engines, (1964-11), p.117
- (24) Pachernegg, S.J., SAE paper 670928(1967)
- (25) Gomei, Y., Trans. Japan Soc. Mech. Engrs., (in Japanese), Vol.38, No.314, (1972-10), p.2641