The Prediction of Volumetric Efficiency Considering Gas Exchange Process in Spark Ignition Engine

전기점화기관에서 흡배기과정을 고려한 체적효율의 예측 및 실험

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요 약

본 논문은 단기통 4사이클 전기점화기관에서 가스교환 과정이 체적효율에 미치는 영향에 대하여 연구한 것이다.

가스교환 과정의 수학적 모델을 설정하고 수치해석을 수행한 결과 체적효율은 가스교환 과정과 관련되는 밸브 개폐시기(1, 2)보다는 흡배기관의 압력에 의하여 더 큰 영향을 받는다는 것을 알 수 있었다.

실험에 있어서는 각각 밸브 개폐시기가 다른 3종류의 캠을 사용하였으며 수치 해석결과와 실험결과가 비교적 잘 일치하였다.

INTRODUCTION

RECENTLY internal combustion engine designers have to face various limitations in view of the environmental hazard and energy conservation. As a result, it is important to understand the fundamental processes in engines. The gas exchange process, one of the essential processes, affects engine output, fuel economy and emission level. However the complexity of this process make it difficult in systematic development of the intake and exhaust pipes. One of the tools that are useful in this field is a simulation analysis (3).

For many years a comprehensive study on the gas dynamic performance of the internal combustion engine has been carried out by R.S. Benson (4,5). Based on this study, further developments (6,7) have been made to closely simulate the real processes.

In the present study, the volumetric efficiency was predicted by considering the gas exchange process for a 4-stroke, single-cyclinder, spark-ignition engine. The mathematical model was formulated to simulate the flow in the intake and exhaust pipes such that the mass flow rate, the pressure in the cylinder and pipes, and the volumetric efficiency were calculated. Experiment was performed in order to conform the above mathematical model. A comparison of

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the calculated results with the experimental data was presented. Finally the effects of com profile on volumetric efficiency were examined.

MATHEMATICAL MODEL

GOVERNING EQUATIONS – During the gas excange process, the situation in the intake and exhaust pipes were assumed to be with homentropic flow without heat loss and friction loss.

This situation of homentropic flow is a special case of the general flow problem and its solution is reasonably compared with the more complicated general flow problem.

The gas flow in the pipes and cylinder is assumed;

1. For the intake and exhaust pipes, the gases obey the ideal gas properties.
2. The gas flow in the pipe was taken as one dimensional unsteady compressible flow and uniform flow with regard to pipe section.
3. In the cylinder, the pressure and temperature are temporally uniform.
4. For all domain, the entropy of gas is constant except the carburetor, intake and exhaust values.
5. The friction loss and heat loss in the pipes are neglected.

The governing equations of the system are; continuity, momentum equazion, and homentropic equation of ideal gases.

$$\frac{\partial p}{\partial t}+\frac{\partial(pu)}{\partial x}=0 \quad \cdots\cdots (1)$$

$$\rho\frac{Du}{Dt}+\frac{\partial p}{\partial x}=0 \quad \cdots\cdots (2)$$

$$a^2=\frac{kp}{\rho} \quad \cdots\cdots (3)$$

From the above equations, the hometropic flow equations are obtained in terms of a and u,

$$\frac{2}{k-1}\frac{\partial a}{\partial t}+\frac{2}{k-1}u\frac{\partial a}{\partial x}+a\frac{\partial u}{\partial x}=0 \cdots\cdots (4)$$

$$\frac{2}{k-1}a\frac{\partial a}{\partial x}+\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}=0 \quad \cdots\cdots (5)$$

These equations were known quasi-linear hyperbolic partial differential equations. There is no analytical solution such that numerical analysis must be adopted.

To obtain a numerical solution the method of characteristics is used. The basic procedure is to transform the two separate equations of the solution, namely, $a=a(x,t)$ and $u=u(x,t)$ to another grouping $c=c(x,t)$ and $c=c(u,a)$ A unique relationship between i) c, u and a is sought so that the solution ii) $c=c(x,t)$ will give numerical values for iii) u and a at x and t. Equation (4) and (5) are represented by the characteristic equations as

$$\frac{dx}{dt}=u\pm a \quad \cdots\cdots (6)$$

$$\frac{da}{du}=\mp\frac{k-1}{2} \quad \cdots\cdots (7)$$

these now become, in dimensionless form

$$\left(\frac{dX}{dZ}\right)_\lambda=U+A=b\lambda-a\beta \quad \cdots\cdots (8)$$

$$\left(\frac{dX}{dZ}\right)_\beta=U-A=a\lambda-b\beta \quad \cdots\cdots (9)$$

where,

$$\lambda=A+\frac{k-1}{2}U,\quad \beta=A-\frac{k-1}{2}U$$

and

$$a=\frac{3-k}{2(k-1)}, \quad b=\frac{k+1}{2(k-1)}$$

The numerical form of λ_I and λ_{II} transformed from λ and β are

$$(\lambda_{I})_{r+1,s} = (\lambda_{I})_{r,s} + \frac{\Delta Z}{\Delta X}\{ b(\lambda_{I})_{r,s-1} - a (\lambda_{II})_{r,s-1}\}\{(\lambda_{I})_{r,s-1} - (\lambda_{I})_{r,s}\} \quad \cdots\cdots (10)$$

$$(\lambda_{II})_{r,s+1,s} = (\lambda_{II})_{r,s} + \frac{\Delta Z}{\Delta X}\{ b(\lambda_{II})_{r,s+1} - a (\lambda_{I})_{r,s+1}\}\{(\lambda_{II})_{r,s+1} - (\lambda_{II})_{r,s}\} \quad \cdots\cdots (11)$$

where the subscript r and s are mech point in $Z-X$ plane.

Then the time step is based on the stability criterion for all mesh points;

$$\frac{\Delta Z}{\Delta X} < \frac{1}{A + |U|} \quad \cdots\cdots (12)$$

Finally, the pressure and velocity can be obtained in terms of λ_I and λ_{II} at the appropriate time intervals and locations.

$$\left(\frac{p}{p_{ref}}\right)^{(k-1)/2k} = \frac{(\lambda_I) + (\lambda_{II})}{2} \quad \cdots\cdots (13)$$

$$\left(\frac{u}{a_{ref}}\right) = \frac{(\lambda_I) - (\lambda_{II})}{k-1} \quad \cdots\cdots (14)$$

BOUNDARY CONDITIONS – The boundary conditions are specified at the positions of the outflow from the cylinder to the valves, and the outflow from the pipes to the partially open ends (nozzle models). Also the flow was considered for both sonic and subsonic depending on the pressure.

The model of the outflow from the cylinder to the valves were used for both the flow from cylinder to the exhaust valve and the reverse flow to the intake valve.

The pressure in the cylinder is obtained as follows.

$$\underset{\dot{m}_a,\ h_{oa},\ a_{oa}}{\text{intake in}} \longrightarrow \boxed{\begin{array}{l}\text{Cylinder}\\ m_c,\ V_c,\ h_c,\ p_c,\ e_c\end{array}} \longrightarrow \underset{\dot{m}_e,\ h_{oc}}{\text{exhaust out}}$$

control volume

The cylinder is taken as a control volume. The generalized first law of thermodynamics for the control volume can be written as;

$$\dot{Q} - \dot{W}_s - \dot{W}_p = \frac{\partial (E)_{cv}}{\partial t} + \dot{m}_{out}(h_o)_{out} - \dot{m}_{in}(h_o)_{in}$$

The rate of pressure rise was obtained from the relation

$$\dot{P}_c = \frac{1}{V_c}\left(a_{oa}^2\ \dot{m}_a - a_c^2\ \dot{m}_e - k\ p_c\ \dot{V}_c\right)$$

The rate of change of cylinder volume was given by

$$\dot{V}_c = F_c\ r\left[\sin\theta + \frac{1}{2}\left\{\frac{\sin 2\theta}{\sqrt{n^2 - \sin^2\theta}}\right\}\right]\ 2\pi N$$

where,

F_c	:	cylinder cross-section area
N	:	crank speed in rev/s
n	:	connecting rod length/crank radius
θ	:	angle from TDC

Nozzle model was applied to the inflow from the intake pipes to the cylinder and the outflow from the exhaust pipes to atmosphere, and the reference pressure outside of the nozzle is a static one.

COMPUTER PROGRAM – Computation was performed by using the cylinder pressure measured when the exhaust valve was opened and the temperature in the cylinder assumed from the engine operating conditions as the initial values. Also the different reference temperatures were assigned at the intake and exhaust pipes, which given different entropy level to each other.

The time step was determined to satisfy the stability criterion and to make it the same as absolute time step at the intake and exhaust pipes. By checking the mass balance between the inflow to the cylinder and the outflow from the cylinder through the values during the time from exhaust valve opening to intake valve closing, the volumetric efficiency was calculated.

EXPERIMENTAL APPARATUS AND PROCEDURE

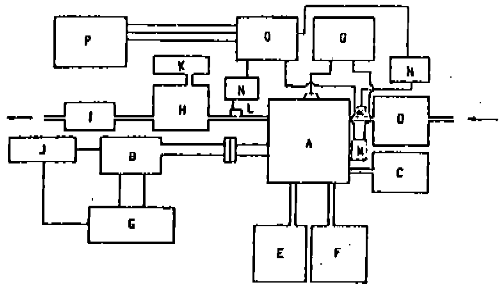
The engine used was a single-cylinder, water-cooled, spark-ignition engine connected with a dynamometer. The fuel supply system for this engine was composed of both the conventional carburetor and the electronic fuel injection system. The spark timing and valve timing can be changed manually. The specifications of this engine are listed in table 1.

Table 1 – Specifications of Test Engine

Number of Cylinders	1
Bore x Stroke (mm)	4
Displacement (1)	85 x 86
Displacemeng (1)	0.488
Compression Ratio	8.5
Combustion Chamber Type	Wedge-type
Valve Mechanism	OHV

To hold the static pressure at the outlet of the exhaust pipe, a surge tank was used, the dimensions of which are 450(L) x 600(W) x 450(H) mm. The pressure in the surge tank was measured by a manometer. The length of the exhaust pipe was 2.4 m, with the diameter of 0.037m, the length of the intake pipe was 0.8m, with the diameter of 0.038m.

The pressure in the intake pipe was measured with a strain gage type pressure transducer (Kyowa Inc. model PG-2KU) located 0.185m from the intake valve. The pressure in the exhaust pipe was measured with a piezo-electric type pressure transducer (PCB Inc. model 101A05) which was mounted 0.28m downstream of the exhaust valve. The cylinder pressure was measured with a pressure transducer Kistler 601A and recorded and analyzed with a combustion analyzer Ono Sokki, CB-366. The Schematic diagram of the experimental system is shown in Fig. 1.



A. single cylinder E/G
B. E/G dynamometer
C. fuel system
D. carburetor
E. cooling system
F. lubricating system
G. load control system
H. exhaust surge tank
I. muffler
J. tachometer
K. manometer
L. ex. press. sensor
M. in. press. sensor
N. amplifier
O. A/D converter
P. microcomputer
Q. combustion analyzer
R. TDC and angle sensor

Fig.1 Schematic diagram of experimental apparatus

To investigate the influence f valve timing on the volumetric efficiency, the experiments were performed for the three models of cams which had different operning and closing timing, as shown in Table 2. The valve clearance was adjusted to have the original design value of the engine.

The intake valve lift and the exhaust valve lift obtained with each came are shown in Figure 5-7 with the respective valve timing and valve overlap (8, 9).

The experiments were performed for every 500 rpm between 1500 to 3500 rpm, and the spark timing was set at MBT for all experiments.

Table 2. Valve timings for 3-models of cams

	EVO	EVC	EX. DUR.	IVO	IVC	IN. DUR.
MODEL 1	BBDC 52°	ATDC 12°	244°	BTDC 16°	ABDC 54°	250°
MODEL 2	BBDC 58°	ATDC 22°	260°	BTDC 10°	ABDC 60°	250°
MODEL 3	BBDC 54°	ATDC 14°	248°	BTDC 14°	ABDC 56°	250°

The measured pressure data in the intake and exhaust pipes were fed to a personal computer. The pressure in the combustion chamber was measured for every one degree whereas that in the intake and exhaust pipes was measured for every two or three degrees of crankangle for each engine speed. The operating conditions are shown in table 3.

Table 3. Operating Conditions

Load	Full Throttle
Speed, rpm	1500 – 3500
Spark Advance	MBT
Fuel	Normal Gasoline
Air-Fuel Ratio	Stoichiometry
Temp. of Cooling Water	80±1°C
Temp. of Lubrication Oil	82±1°C
Inlet Mixture Temp.	25±2°C

RESULTS AND DISCUSSIONS

Figures 2 and 3 show the intake and exhaust pressures obtained from the experiment and the calculation for engine speed of 2000 and 2500 rpm, respectively.

The period of pressure waves in the exhaust pipe decreases as the engine speed increases. It is about 130 degrees (0.0108 s) of crankangle at 2000 rpm and 150 degrees (0.01 s) at 2500 rpm.. The calculated results show that, at 2500 rpm, the period changes from 140 degrees (crankangle) to 155 degrees

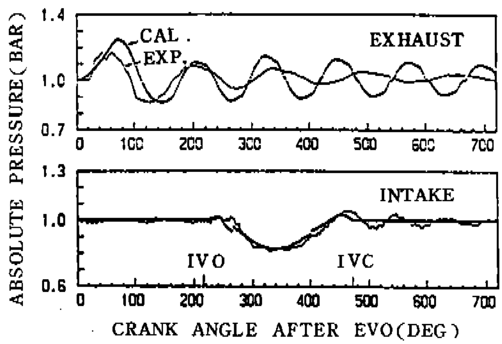


Fig.2 Comparison of the pressure variation in the intake and exhaust pippes for the experimental data the calculated data at 100% open throttle, 2000 rpm, model 1

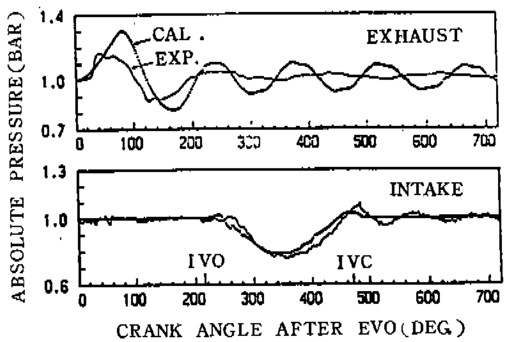


Fig.3 Comparison of the pressure variation in the intake and exhaust pipes for the experimentsl data and the calculated data at 100% open throttle, 2500 rpm, model 1

as the length of th exahaust pipe increases from 1.6m to 3.0m for a given intake pipe length of 0.8m. It appears from the experimental results that the amplitude of the pressure wave attenuates rapidly and completely dies out at 720 degrees. But the calculated results show rater slow decrease compared with the experimental data. This diference results from the neglection of the heat loss and the friction loss in calculation. The pressure in the exhaust pipe changes about 0.03 to 0.12 bar (20 to 90 mmHg) with the variation of operating conditions because the static pressure condition at the outlet of the exhaust pipe is not satisfied.

As the engine speed increases the period of pressure wave decreases, which can be explained as follows. Once the pressure wave is formed it travels to the outlet of the exhaust pipe with the velocity of the sonic velocity a plus the gas flow velocity u. It reflects at the outlet of the exhaust pipe and returns to the exhaust valve with the velocity of the sonic velocity a minus the gas flow velocity u, and as the engine speed increases, the exhaust gas temperature and the gas flow velocity increase.

With above phenomena, if the backward pressure wave arrives at the exhaust valve when it opens, the backpressure of the engine decreases. Thus the output power of the engine and the volumetric efficiency increase. As the backpressure decreases, the density of residual gas in clearance volume decreases. And, if the intake valve is opened, the scavenging effect of the residual gas can be expected.

Both the experimental results and calculated data indicate a pressure rise in the intake pipe of about 0.02 bar during 30 degrees (crankangle) after intake valve opening. It can be explained with the aid of Fig. 4 showing the vriation of pressure in the cylinder. Near TDC, the cylinder pressure increases. The rate of the volume change due to the piston movement is large compared with the gas flow rate through the exhaust valve, which is the reason for the pressure rise in the intake pipe. The pressure rise also accompanies backflow of the gas from the combustion chamber into the intake pipe.

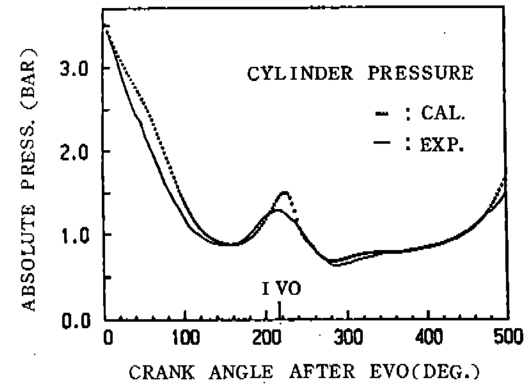


Fig.4 Comparison of the cylinder pressure for the experimental data and the calculated at 100% open throttle, 2500 rpm, model 1

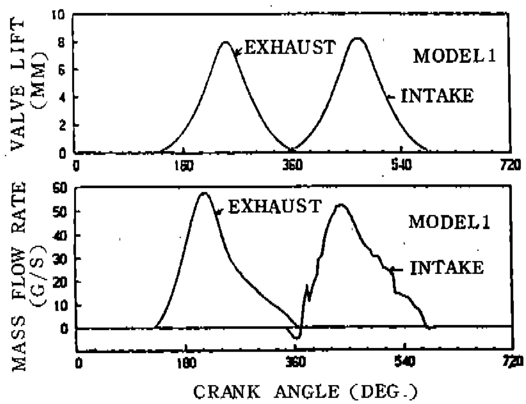


Fig.5 Valve displacements and mass flow rates for the intake and exhaust valves at 100% open throttle, 2500 rpm, model 1

The mass flow rate through each valve is plotted together with the valve lift and the engine speed for model 1, 2, and 3, respectively in Figures 5, 6 and 7. It is apparent from the figures that the exhaust mass flow rate shows a smooth curve, whereas the intake mass flow rate indicates a rather irregular trend because of the inertia effect. The

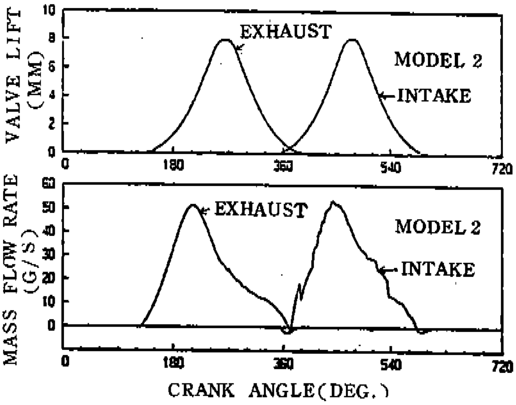


Fig.6 Valve displacements and mass flow rates for the intake and exhaust valves at 100% open throttle, 2500 rpm, model 2

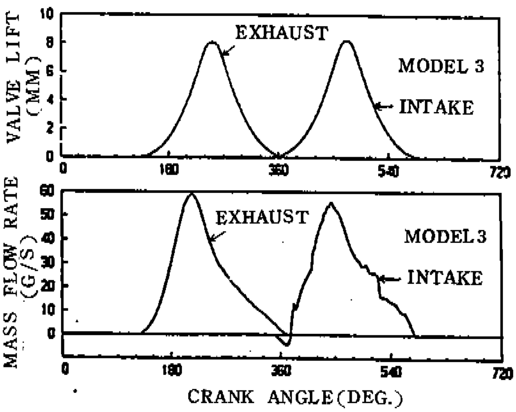


Fig.7 Valve displacements and mass flow rates for the intake and exhaust valves at 100% open throttle, 2500 rpm, model 3

intake mass flow rate shows its maximum value about 80 degrees (crank angle) after intake valve opening for all models. The exhaust mass flow rate shows the maximum value at 90, 95 and 90 degrees (crankangle) after exhaust valve opening for model 1, 2, and 3, respectively. The intake mass flow rate represents the maximum for 45 degrees before the maximum lift point and the exhaust mass flow rate indicates the maximum for 26-35 degrees before the maximum lift point.

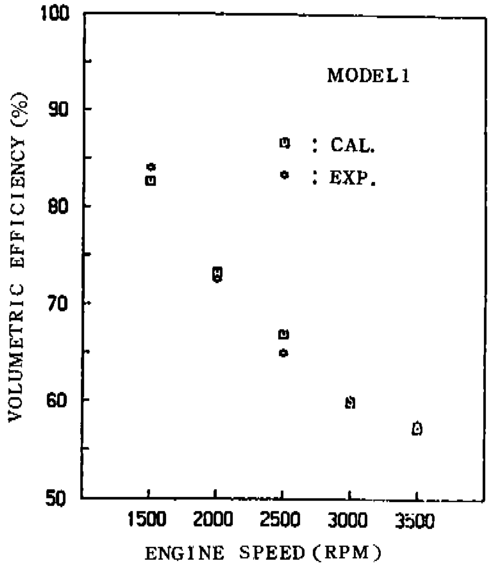


Fig.8 Comparison of the volumetric efficiency for the experimental and calculated data along the engine speed at 100% open throthle, model 1

Also the backflow phenomena toward the intake pipe occur during 30 degrees after intake valve openin for all models.

Figures 8, 9, and 10 show the experimental data and computed results for the volumetric efficiency. The results show a satisfactory agreement between the prediction and the experimental data for a wide range of operating conditions. Since the cam models used are similar, the mass flow rate and the volumetric efficiency represent 2-3% of difference. However it is found that the trend of volumetric efficiency changes with the engine speed.

CONCLUSIONS

For thedetailed predictions of the gas exchange process, stricter assumption than the homentropicity is required. However based on the comparisons with the experi-

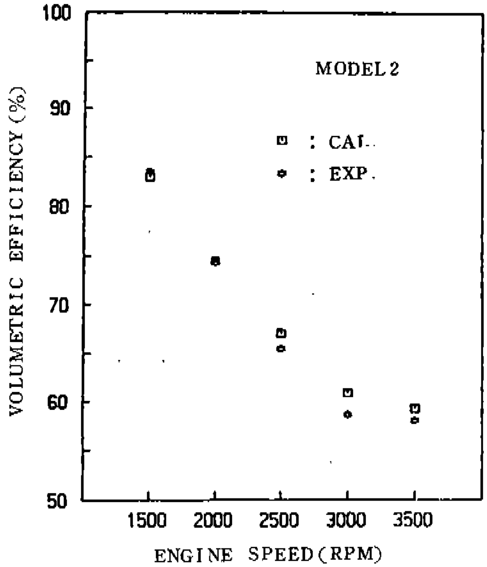


Fig.9 Comparison of the volumetric efficiency for the experimental and calculated data along the engine speed at 100% open throttle, model 2

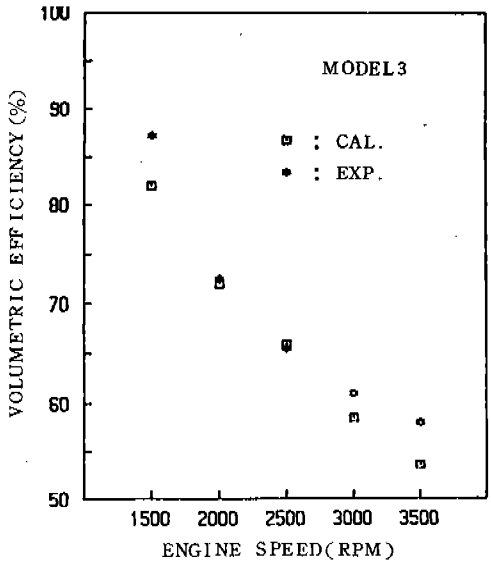


Fig. 10 Comparison of the volumetric efficiency for the experimental and calculated data along the engine speed at 100% open throttle, model 3

mental data, the mathematical model of the gas exchange process can reasonably predict the volumetric efficiency and other relevant parameters. The comparisons of the calculated and experimental results are summarized as follows.

1. The complex phenomena of gas exchange process in the intake and exhaust pipes can be explained with a simplified mathematical model by adjusting the ratio of nozzle area and the reference temperature.
2. The pressure in the intake pipe is in good agreement, whereas differences of up 7% in period of the pressure wave in the exhaust pipe is observed.
3. The backflow phenomena toward the intake pipe occur during 30 degrees after intake valve opening for all models.
4. The difference in volumetric efficiency is within 5% in the range of 1500-3500 rpm. Also the volumetric efficiency does not show much differences with three cams tested.

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