A Numerical and Experimental Study on the Optimal Design for the Intake System of the MPI Spark Ignition Engines

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A computer program has been developed to predict the engine performance characteristics through the analysis of the flow in the intake and exhaust systems and of the cylinder combustion phenomena for the MPI spark ignition engines. Using the program, a study for the optimal design of the intake system has been performed by varying the factors which can influence the volumetric efficiency, such as the volume of the plenum chamber, the length of the intake manifold and the pipe length between the surge tank and the plenum chamber. Experimental tests have also been carried out to obtain the transient pressure history in the intake manifold and the cylinder, and the variations of volumetric efficiency over the various engine speeds. The result of simulation has been compared with that of experimental test, and the optimal design data for the test engine could be found. The comparison of volumetric efficiency shows good agreement between the simulation and experiment.

Key Words: Volumetric Efficiency, Method of Characteristics, Intake System, Simulation

Nomenclature

a : Speed of sound
a_q : Annand constant for convective heat transfer
A_f : Area of the flame front
b : Annand constant for convective heat transfer
c : Annand constant for radiative heat transfer
C_p : Constant pressure specific heat
C_v : Constant volume specific heat
D : Diameter
f : Friction factor
ff : Turbulent flame factor
F : Area
k : Specific heat ratio
m : Mass
N : Engine speed
P : Pressure
q : Rate of heat transfer per unit mass
Q : Total heat flux
R : Gas constant
R_e : Reynolds number
t : Time
T : Temperature
u : Velocity or internal energy
u_l : Laminar flame speed
u_t : Turbulent flame speed
V : Volume
x : Distance
α : Crank angle
ρ : Density

suffices

c : Cylinder
e : Exhaust
i : Intake
m : Unburnt mixture
o : Stagnation state
p : Pipe or combustion product
w : Wall

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1. Introduction

Pressure waves of the induced charge are generated periodically in the intake system when the fresh charge is induced into the cylinder intermittently in internal combustion engines. It is well known that the volumetric efficiency can be improved when the dynamic effect of wave is properly used.

The dynamic effect is generally classified into two parts: the inertia effect and pulsation effect. The expansion waves developed at the openning of intake valve propagate toward the other end of the intake pipe. These expansion waves may be reflected as positive pressure waves at the end of manifold (or at the plenum), and propagated back to the cylinder end of the pipe. If the waves propagated back and forth between the intake valve and the open end make influences on the next cycle, then it is called "the pulsation effect", and if they have influence on the same cycle, it is called "the inertia effect".

Since the mode of inertia supercharging is very complicated and takes hours of costly downtime for experiment, when the optimal intake system is to be developed, the time consuming efforts or expenses can be saved by the effective simulation models. There are several numerical methods which can predict the unsteady behavior of the gas flow in the intake and exhaust systems. The finite difference method (Lee, 1989) and the method of characteristics (Benson, 1982 ; Benson, 1964) are the typical tools among the quantitative methods. The amount of induced air can be calculated effectively through one of these methods.

This article provides the numerical and experimental study for the optimal design of the intake system of spark ignition engines. In this study, the method of characteristics and two zone combustion model are employed to calculate gas properties in the pipes and cylinder. The experiments for the verification of the simulation models are also accomplished over the various engine speeds. The transient pressure variations in the cylinder and the intake manifold are obtained from the data acquisition system. On the basis of the comparison of pressures through simulations and experiments, the analytical model is then modified.

After investigating the validity of the simulation models, the optimal geometry of the intake system has been obtained from simulation prior to experiment. The volumetric efficiency of each intake system was examined with varying volume of the plenum chamber, length of manifold between the cylinder and plenum chamber, and length of pipe between the plenum chamber and open end of pipe. And the optimal geometry with the best volumetric efficiency was then determined. Finally, experiments were performed over the range in the neighborhood of the optimal geometry obtained from the simulation. The comparison of volumetric efficiency shows good agreement between the simulation and the experiment.

2. Simulation Models

In order to predict the volumetric efficiencies over the various configurations of the intake system, which is the key point of this article, the changes of properties at the whole components composing the engine system including the intake system, the exhaust system and cylinder are considered. The models adopted in this research are briefly described in the following sections.

2.1 Cylinder properties

The properties in the cylinder are often calculated using two models, single zone model and two zone model, according to the distribution of the cylinder contents.

Single Zone Model - The cylinder contents are considered to be uniformly distributed throughout the cylinder for the compression of the mixture of the fresh air and residuals, and the expansion of the products after combustion, and the gas exchange process (Benson, 1975).

With the aid of the first law of thermodynamics, the change of cylinder pressure with crank angle is derived as:

$$\frac{dP_c}{d\alpha}=\frac{k_c-1}{V_c}\left[-\frac{k_c}{k_c-1}P_c\frac{dV_c}{d\alpha}+\sum\frac{a_{io}^2}{k_i-1}\frac{dm_i}{d\alpha}-\sum\frac{a_{ie}^2}{k_e-1}\frac{dm_e}{d\alpha}+\frac{dQ}{d\alpha}\right] \quad (1)$$

The convective and radiative heat transfer rate from gas to the wall, $dQ/d\alpha$, is calculated by using the Annand equation (Annand, 1963) :

$$\frac{dQ_c}{d\alpha}=\frac{F}{6N}\frac{a_q R_e^{\,b} k}{D_c}(T_w-T)+c(T_w^4-T^4) \tag{2}$$

The change of mass in the cylinder is determined from the continuity equation:

$$\frac{dm_c}{d\alpha}=\sum\frac{dm_i}{d\alpha}-\sum\frac{dm_e}{d\alpha} \tag{3}$$

where the mass flow rates entering and leaving the cylinder, $dm_i/d\alpha$ and $dm_e/d\alpha$, are determined from the pipe flow calculations, and the summation notations are for multi-valve calculation. The pressure and mass in the cylinder are determined by the simple time integration of Eqs. (1) and (3). The gas temperature in the cylinder is evaluated from the equation of state.

Two Zone Model - After the ignition of the air-fuel mixtures, the cylinder is assumed to be divided into two zones, burnt and unburnt zones, by spherical flame front. In the two-zone model (Benson, 1975), it is assumed that the pressure is uniform throughout the cylinder while the temperatures, specific heats and chemical compositions differ in each zone. In the model, the mass burning rate, $dm_p/d\alpha$, is represented as:

$$\frac{dm_p}{d\alpha}=\rho_m A_f u_t \tag{4}$$

where u_t and A_f are the turbulent flame speed and the area of the flame front, respectively.

The turbulent flame speed is given by the product of the turbulent flame factor, ff, and the laminar flame speed, u_l, suggested by Kuehl (Kuehl, 1962) as:

$$u_t=ff u_l=ff\frac{(0.78\times10^4)P^{-0.09876}}{\left[\frac{10000}{T_p}+\frac{900}{T_m}\right]^{4.938}} \tag{5}$$

The turbulent flame factor is dependent on the engine type and operating conditions. In this study, the turbulent flame factor, which is based on the turbulent intensity, suggested by Lienesch (Lienesch, 1980) is used. The combustion period is divided into four stages: stage of the ignition delay, stage of developing flame, stage of fully developed flame, and stage of flame decaying. The turbulent flame factor is differently expressed for each stage.

With the mass burning rate, the rates of change for burnt and unburnt gas temperatures and cylinder pressure are calculated as:

$$\frac{dP}{d\alpha}=\left[(1+\frac{c_{vp}}{R_p})P\frac{dV}{d\alpha}+[(u_p-u_m)-c_{vp}(T_p-\frac{R_m T_m}{R_p})]\frac{dm_p}{d\alpha}+\left(\frac{c_{vm}}{c_{pm}}-\frac{c_{vp}}{R_p}\frac{R_m}{c_{pm}}\right)\frac{dQ_m}{d\alpha}-\frac{dQ}{d\alpha}\right]\Bigg/\left[-\left[\frac{c_{vm}}{c_{pm}}V_m-\frac{c_{vp}}{R_p}\frac{R_m}{c_{pm}}V_m+\frac{c_{vp}}{R_p}V\right]\right] \tag{6}$$

$$\frac{dT_m}{d\alpha}=\frac{V_m}{m_m c_{pm}}\frac{dP}{d\alpha}+\frac{1}{m_m c_{pm}}\frac{dQ_m}{d\alpha} \tag{7}$$

$$\frac{dT_p}{d\alpha}=\frac{P}{m_p R_p}\left[\frac{dV}{d\alpha}-\left(\frac{R_p T_p}{P}-R_m\frac{T_m}{P}\right)\right.$$

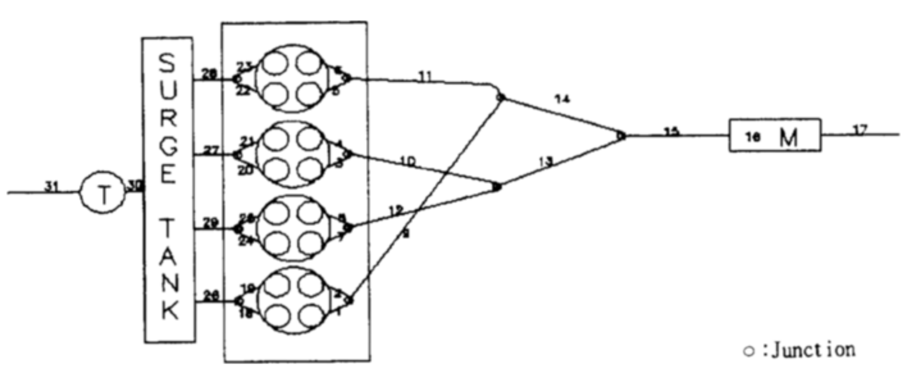

Fig. 1 Layout of test engine

$$\frac{dm_p}{d\alpha} - \frac{R_m V_m}{P c_{pm}} \frac{dP}{d\alpha} - \frac{R_m}{P c_{pm}} \frac{dQ_m}{d\alpha} + \frac{V}{P} \frac{dP}{d\alpha} \Big] \tag{8}$$

The assumptions in modeling the in-cylinder processes may produce an error in predicting the pressure history in the cylinder. Although the error in pressure history may lead to the other errors in predicting the indicated mean effective pressure, it has little influence to the prediction of volumetric efficiency.

2.2 Simulation of intake and exhaust systems

The analysis of the flow in the intake and exhaust systems includes not only the flow in the pipes but also in the boundaries, such as the intake and exhaust valves, throttle body, manifold junctions, plenum chamber, mufflers and open ends. The layout for the intake and exhaust systems of test engine is shown in Fig. 1.

Pipe Flow Calculation - The flow in the intake and exhaust systems is regarded as the one dimensional unsteady flow including wall friction, gradual area change, heat transfer and entropy gradients. The basic equations are

- Continuity Equation :

$$\frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial x} + u \frac{\partial \rho}{\partial x} + \frac{u}{F} \frac{dF}{dx} = 0 \tag{9}$$

- Momentum Equation :

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{1}{\rho} \frac{\partial P}{\partial x} + \frac{4f}{D} \frac{u^2}{2} \frac{u}{|u|} = 0 \tag{10}$$

- Energy Equation :

$$q \rho F dx = \frac{\partial}{\partial t} \Big[(\rho F dx) \Big(C_v T + \frac{u^2}{2} \Big) \Big] + \frac{\partial}{\partial x} \Big[\rho u F \Big(C_v T + \frac{P}{\rho} + \frac{u^2}{2} \Big) \Big] dx \tag{11}$$

Equations (6) through (11) are a system of quasi-linear hyperbolic partial differential equations. Using the method of characteristics (Benson, 1982 ; Benson, 1964 ; Lee, 1991), the governing partial differential equations are transformed into ordinary differential equations along the path lines and wave characteristics. These equations are solved numerically by using the mesh method for wave characteristics and the modified non-mesh method (Benson, 1982 ; Benson, 1964) for the path lines to enable the calculation of properties at required locations and times. The gas compositions are calculated along the path lines. In addition, the variation of specific heat due to the temperature and composition is also considered.

Boundary Conditions- The models used in analyzing the boundary components shown in Fig. 1 are briefly described as:

(1) Flow through Valves --- The flow through intake or exhaust valves is analyzed by the constant pressure model (Benson, 1982 ; Daneshyar, 1968). In the model, for the flow out of cylinder, the pressure at valve throat is regarded as being identical to the pipe pressure for the subsonic flow, whereas the pressure drop is considered when it is choked in the throat. For the flow into the cylinder, the gas undergoes isentropic flow from the pipe to the valve throat, and the pressure at the throat is assumed to be equal to that in cylinder.

(2) Flow through Throttle Body --- For the analysis of the flow across throttle body, the adiabatic pressure loss model (Benson, 1982 ; Benson, 1974) is used. The throttle body is considered as a discontinuity across which the pressure drops adiabatically. The resistance coefficient determined empirically for each throttle setting is required to evaluate the effect of the pressure drop across the throttle body (Yoo, 1991).

(3) Manifold Junction --- The effect of the interference by waves transmitted from adjacent cylinders through manifold junctions in the multi-cylinder engine is evaluated by the constant-pressure perfect-mixing model (Benson, 1982 ; Benson, 1975). In the model, the pressures at pipe ends at the junction are identical and the gases leaving the junction have the weighted mean values of concentrations of the gas species entering the junction.

(4) Muffler --- The muffler is regarded as a

pipe with large diameter with sudden area changes i. e., one side of the pipe is sudden enlargement, the other side sudden contraction (Benson, 1963). The parts inside the muffler are considered by assigning larger friction factor than other pipes.

(5) Open End to Atmosphere --- When the gas inside the pipe flows into the atmosphere, the pressure at the pipe end is assumed to be the same as the atmospheric pressure. For the case that the atmospheric air flows into pipe, the gas properties are calculated by applying the energy equation between the inside and outside of the pipe (Benson, 1982).

(6) Plenum Chamber --- The rates of change of pressure and mass in plenum chamber are determined by Eqs. (1) and (3) with setting $dV/d\alpha=0$. The summation is for all of the pipes connected to the plenum chamber.

3. Experiments

In order to investigate the validity of the computer program developed, the measurement of the mass flow rates of air and fuel and the transient pressures in the cylinder and the intake pipe were conducted. The test engine is a 4-cylinder, 4-cycle multi-point injection spark ignition engine and its specification is listed in Table 1.

Table 1 Engine specification

Engine type		4-Cycle, 4-Cylinder In-line, DOHC SI engine
Displacement volume		1836cc
Bore × Stroke		81.5×88.0mm
Compression ratio		9.2
Valve timing	IVO	26° BTDC
	IVC	46° ABDC
	EVO	55° BBDC
	EVC	9° ATDC
Maximum output		135 PS/6000rpm
Maximum torque		17.5kg m/4500rpm
Plenum chamber		1100cc (Original)

A piezoelectric pressure transducer (KISTLER 6117A17) with a spark plug adaptor was employed to measure the transient cylinder pressure. For the transient intake pressure variation with crank angle, a piezoresistive pressure transducer (KISTLER 4075A10) which can measure the absolute pressure variation precisely, was installed at the intake manifold connected to the first cylinder. The pressure signals were processed by the data acquisition system with the crank angle pulses detected by rotary encoder.

In order to obtain an optimal intake system, components of intake system were made to be changeable. Figure 2 shows a schematic diagram of the intake system. Tests were conducted for the volumes of the plenum chamber of 1200, 2000 and 3000cc. The length of intake pipe was employed from 100cm to 220cm in increments of 40cm. The length of intake manifold was examined for the cases of 30cm, 60cm and 90cm. For each system, the volumetric efficiencies were measured for the engine speed of 2000 to 3500 rpm in increments of 500 rpm.

4. Results and Discussions

4.1 Transient pressure variations

The numerical and experimental pressure curves in the cylinder and the intake manifold are shown in Fig. 3 and Fig. 4 respectively. There is a little discrepancy between numerical and experimental values in the pressure curves, but their tendency of variation agrees well qualitatively. The errors in cylinder pressure curve are observed at early period of combustion, which suggests the necessity of the modification of the combustion modeling. Since the pressure differences between the pipes at manifold junction are neglected, the calculated pressure at the manifold is shown to be higher than the experimental one as shown in Fig. 4. In addition, it is shown that the pressure in manifold at the time of intake valve closure (IVC)

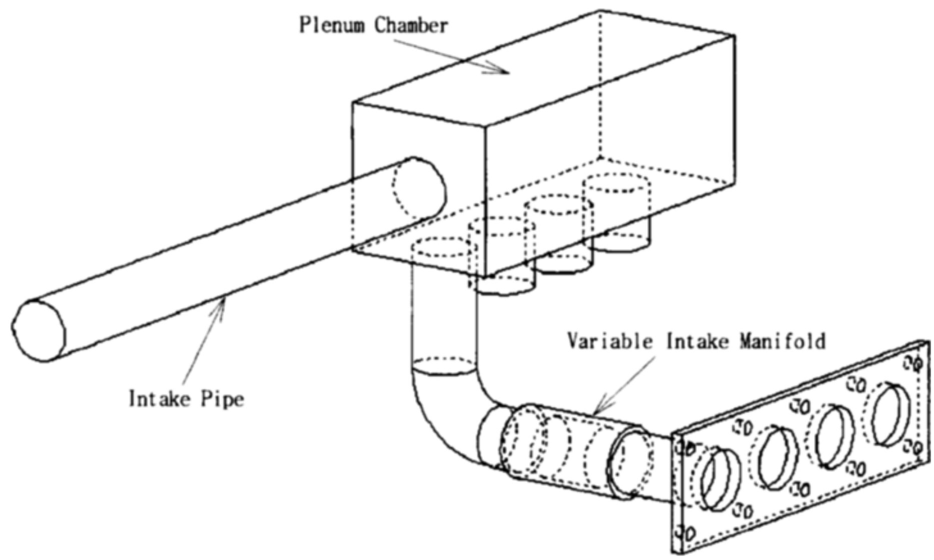

Fig. 2 Schematic diagram of the intake system

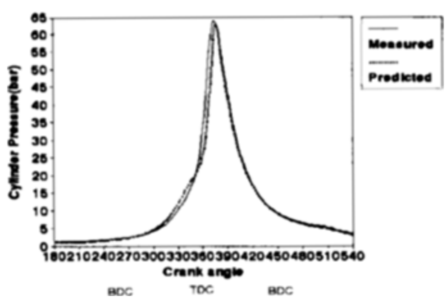

Fig. 3 Pressure variation with crank angle in the cylinder(manifold length 90cm, 3000 rpm WOT)

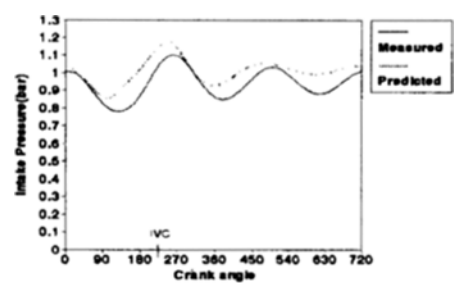

Fig. 4 Pressure variation with crank angle in the intake manifold(manifold lenght 90cm, 3000 rpm WOT)

reaches above 1 bar. This implies that the intake system is tuned well. Through the comparisons, the simulation program is proved to be useful enough for analysing the intake system of the SI engines.

4.2 Intake pipe length

Using the simulation program, the volumetric efficiencies are calculated for the length of the intake pipe from 100 to 220cm under various engine speeds. The results are given in Fig. 5. The maximum values of the volumetric efficiency occur when the length of the intake pipe is 140cm. Therefore, three representative lengths of the intake pipes including 140cm are chosen for experiment keeping the designed volume of the plenum chamber, 1100cc, and the designed length of mainfold, 35cm. Figure 6 shows that the volumetric efficiencies in experiments are best around the length of 140cm as those in simulation.

4.3 Plenum chamber volume

Volumetric efficiencies are calculated for the plenum chamber volume from 1200 to 4000cc at each engine speed of 2000 to 3500 rpm. The maximum value of volumetric efficiency is found at 2000cc of the plenum chamber volume from the simulation results shown in Fig. 7. Therefore, three sets of the intake system with plenum chamber volume of 1200, 2000 and 3000cc are prepared for experiments. Note that the length of the intake pipe is kept as fixed value of 140cm. Figure 8 shows that the plenum chamber volume of 2000cc gets the best experimental volumetric efficiency. Therefore, the value of 2000cc is presumed to be the optimum volume of plenum chamber.

4.4 Manifold length

Figure 9 shows the simulated results of the volumetric efficiencies with the variation of the intake manifold length keeping optimal values for the volume of plenum chamber and the length of intake pipe chosen before. The maximum value is found at the length of 90cm. Therefore, three lengths of mainfold are selected as 30, 60 and 90 cm for comparative experiments. The maximum experimental volumetric efficiency, shown in Fig. 10, is obtained at the length of 90cm which is also the optimum length of the intake manifold in simulation.

Figure 11 shows that the greatest intake pressure at IVC occurs at the case of the length of 90 cm, which supports the best volumetric efficiency at the length of 90cm.

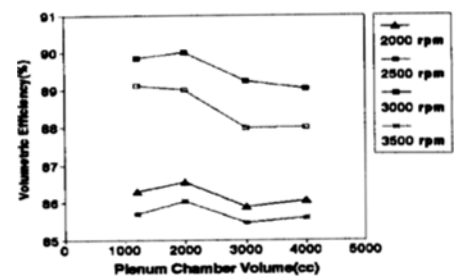

Fig. 7 The effect of the plenum chamber volume on the volumetric efficiency (calculated)

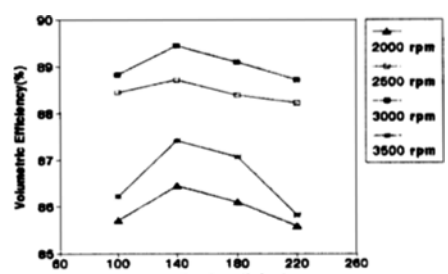

Fig. 5 The effect of the intake pipe length on the volumetric efficiency (calculated)

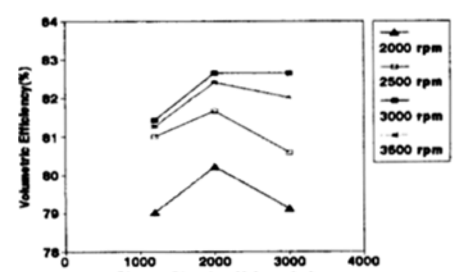

Fig. 8 The effect of the plenum chamber volume on the volumetric efficiency (experiment)

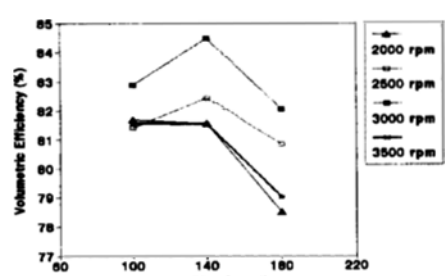

Fig. 6 The effect of the intake pipe length on the volumetric efficiency (experiment)

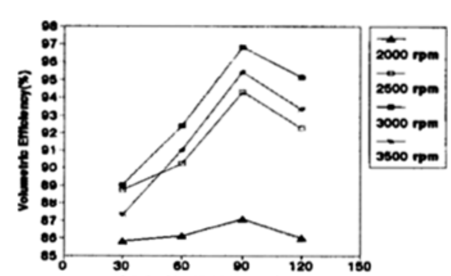

Fig. 9 The effect of the intake manifold length on the volumetric efficiency (calculated)

4.5 The comparison of volumetric efficiencies

Through the simulation and experimental study, the optimum length of the intake pipe and manifold, and the optimum plenum volume are determined as 140cm, 90cm and 2000cc respectively.

Figure 12 shows that the volumetric efficiency of the optimum system obtained through this study is about 4 percent higher than that of the original intake system.

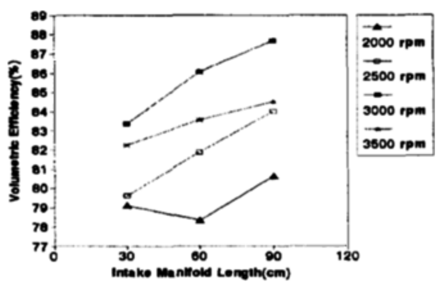

Fig. 10 The effect of the intake manifold length on the volumetric efficiency (experiment)

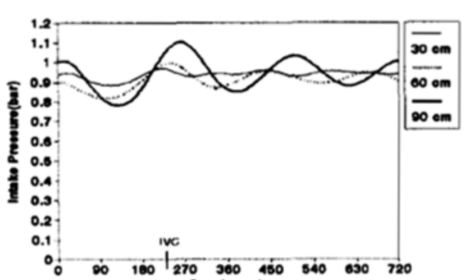

Fig. 11 Comparison of the intake manifold pressure

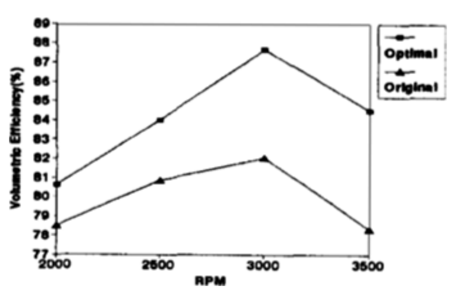

Fig. 12 Comparison of volumetric efficiencies between the optimum intake system and the original system

5. Conclusions

A simulation program has been developed for the prediction of volumetric efficiency of MPI spark ignition engine, and a numerical and experimental study has been carried out, by varying the geometry of the intake system such as the volume of the plenum chamber, the length of intake pipe and intake manifold. The conclusions from both experimental tests and analyses are as follows.

(1) The volumetric efficiencies obtained using the simulation program agree well with the experimental results quantitatively as well as qualitatively.

(2) The optimum dimensions for the intake system of the test engine are determined as: 140cm for the length of the intake pipe, 2000cc for the volume of the plenum chamber and 90cm for the length of the intake manifold.

(3) The volumetric efficiency of the engine with the optimum intake system obtained through this study is about 4 percent higher than that of the original system.

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