Computational and Experimental Analysis of Variable Exhaust Pipe Diameters in Four-Stroke Gasoline Engine

Seuk Cheun Choi †, Hae Jeong Lee*, You Sik Shin*,
Han Shik Chung**, Hyo Min Jeong** and Kwang Young Lee**

4 행정 가솔린 엔진 내의 다양한 배기 파이프 직경 변화에 따른 실험과 수치해석

최석천†, 이해종*, 신유식*, 정한식**, 정효민**, 이광영**

Key Words : Compressible Flow, Pulsating Pressure, Exhaust Pipe, Compression Wave

Abstract

In this study, a experimental method has been introduced for the various exhaust pipe geometry of 4-stroke single cylinder engine. The main experimental parameters are the variation of exhaust pipe diameters and lengths, to measuring the pulsating flow when the intake and exhaust valves are working. As the results of experimental test, the various exhaust geometry were influenced strongly on the exhaust pressure. As the exhaust pipe diameter was decreased, the amplitude and the number of compression wave in exhaust pressure was increased. According to decreasing pipe diameter, the number of compression wave in exhaust pressure was decreased. When the pipe diameter was increase, the second amplitude was increased.

NOMENCLATURE

A	: non-dimensional speed of sound[a/a_{ref}]
a_{ref}	: reference speed of sound[m/s]
bdc	: bottom dead center
d	: diameter of pipe[m]
evo	: exhaust valve open
evc	: exhaust valve close
F	: cross sectional area of pipe[m^2]
ivo	: intake air valve open
ivc	: intake air valve close
N	: engine revolution[rpm]
P	: pressure of pipe[bar]
tdc	: top dead center
U	: non-dimensional velocity[u/a_{ref}]
u	: gas velocity[m/s]
β	: Rimann variable of characteristic
λ	: Rimann variable of characteristic
θ	: crank angle[°,degree]
κ	: ratio of specific heats [C_p/C_v]
ρ	: density[kg/m^3]

SUBSCRIPTS

c	: cylinder condition
cr	: critical condition
in	: characteristic towards boundary
out	: characteristic from boundary
t	: throat condition
1	: intake pipe
2	: exhaust pipe

† Department of Mechanical and Precision Engineering,
The Institude of Marine Industry, Gyeongsang National University
E-mail : chsch300@hotmail.com
TEL : (055)646-4766 FAX : (055)640-3188
* Department of Mechanical and Precision Engineering,
The Institude of Marine Industry, Gyeongsang National University
** School of Mechanical and Aerospace Engineering,
The Institude of Marine Industry, Gyeongsang National University

1. INTRODUCTION

Recently, automotive emissions regulations are more severe by the Environmental Protection Agency (EPA), and many of researchers have a concern to acoustic performance for the intake and exhaust system. The predicting radiated noise requires of the acoustic behavior of the intake and exhaust system and model of the engine cycle source characteristics. This is often dealt with in either the frequency domain or in the time domain analysis. The frequency domain analysis of mufflers is done by means of the transfer matrix method. The time domain modeling by means of characteristics is complete in itself. But this model has some disadvantageous for the aero acoustic performance evaluation of the exhaust system The research about new hybrid approach for predicting of noise radiation from engine exhaust system was suggested by Munjal(1). Also, the three point method was proved that the advantage of analytical simplicity and computational speed, and is as accurate as the boundary element method by Chu et., al..(2)

The use of computer simulation techniques for the improvement of engine development processes has been rapidly expanding over the last decade. Computational methods in fact not only offer reliable predictions of the effect of geometrical modification on engine performance, thus reducing the number of prototypes and experimental tests required for engine optimization, but also allow a better understanding of the unsteady flow phenomena that occur during the gas exchange processes.(3-4) Moreover these techniques provide information on physical quantities which are quite difficult to measure, such as residual exhaust fraction, instantaneous exhaust and intake mass flow rates, etc.

The aim of this paper is therefore to provide an example of the integrated use of computational and experimental techniques for the development of a high performance four-stroke motorcycle engine.

2. METHOD OF NUMERICAL ANALYSIS AND MODEL

2.1 2.1 Method of numerical analysis

The one dimensional and unsteady compressive flow was assumed and the method of characteristic (MOC) was introduced for calculation in each side of intake and exhaust manifold. As the homentropic flow was assumed, the friction and heat transfer was not considered in this study, and the governing equations are constituted with of continuity, momentum and the characteristics that λ and β are given as followings:

$$\frac{\partial\rho}{\partial t}+\rho\frac{\partial u}{\partial x}+u\frac{\partial\rho}{\partial x}=0 \tag{1}$$

$$\frac{\partial u}{\partial t}+u\frac{\partial\rho}{\partial x}+\frac{1}{\rho}\cdot\frac{\partial P}{\partial x}=0 \tag{2}$$

$$\lambda : \frac{dx}{dt}=u+a \ , \quad \frac{du}{dt}=-\frac{\kappa-1}{2} \tag{3}$$

$$\beta : \frac{dx}{dt}=u-a \ , \quad \frac{du}{dt}=\frac{\kappa-1}{2} \tag{4}$$

2.2 Boundary conditions

The length of intake and exhaust pipe were measured by real motor cycle that was selected for standard numerical calculation model. The exhaust pipe diameter of the real motor cycle was $d = 22\,mm$. The boundary condition of intake and exhaust manifold for single cylinder 4-cycle engine are explained as the followings, and the method of characteristics was introduced for numerical analysis, and the main boundary conditions can be written as:

Close end:

$$\lambda_{in}=\lambda_{out} \tag{5}$$

Open end:

$$\lambda_{out}=2-\lambda_{in} \tag{6}$$

Subsonic flow:

$$U^2=\left(\frac{2}{K-1}\right)^2(\lambda_{in}-A) \tag{7}$$

Sonic flow

$$\frac{U}{A}=\phi\left(\frac{A_t}{A}\right)_{cr}^{\kappa+1/\kappa-1} \tag{8}$$

2.3 Target engine and the calculation model of exhaust pipe

The specifications of single cylinder 4-cycle engine are shown in Table 1.

Table 1 Specifications of engine model.

engine type	4-stroke, single cylinder
engine displacement (cc)	124.1
bore (mm) / storoke (mm)	56.5 /49.5
connecting rod length (m)	100
intake valve open (°)	BTDC 10
intake valve close (°)	ABDC 30
exhaust valve open (°)	BBDC 41
exhaust valve close (°)	ATDC 1
compression ratio	9.5 : 1

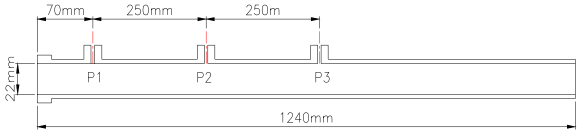

Figure 1. Experimental and numerical result points in exhaust pipe.

Figure 1 shows the experimental and numerical result points in exhaust pipe. The basic length of intake and exhaust pipe was selected from a real motor cycle engine, the exhaust pipe length and diameter were l = 1,240 mm and d = 22 mm, which named standard type of exhaust pipe. The calculation mesh is a constant space grid and the numbers of mesh are one hundred. The temperature and pressure of ambient condition in numerical calculation model are 20 °C and 1 bar, respectively.

Figure 2 shows the intake and exhaust valve opening area according to crank angle. This data was obtained by measurement valve lift. The intake valve opening timing is 138 ° and the exhaust valve opening timing is 348 °. The maximum value of intake valve area have wider than the exhaust valve, and the difference was the maximum intake value area has 21% wider than the maximum exhaust valve area.

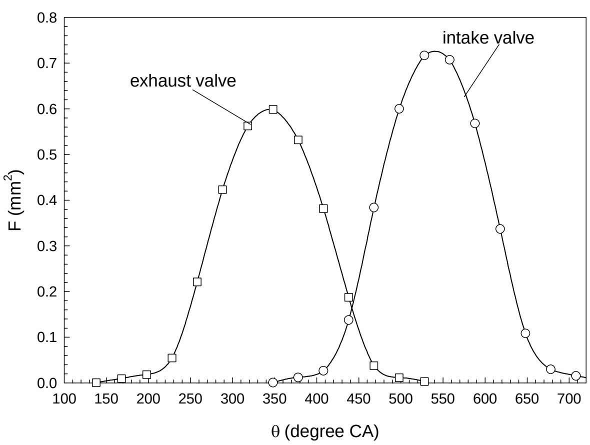

Figure 2. Intake and exhaust valve opening area according to crank angle.

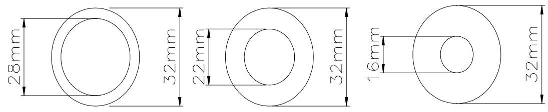

Figure 3. Schematic diagram of various exhaust pipe geometries.

Figure 3 shows various exhaust pipe geometries. The standard type of exhaust pipe is d = 22 mm, a expanded pipe diameter was selected with d = 28 mm and a reduced pipe diameter was selected with d = 16 mm. The basic length of intake and exhaust pipe was selected from a real motor cycle engine, the exhaust pipe length and diameter were l = 1,240 mm and d = 22 mm, which is the case of standard type for exhaust pipe.

3. EPERIMANTAL METHOD

3.1 Experimental procedure

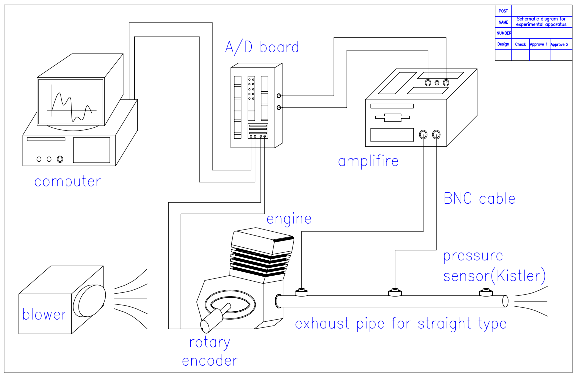

Figure 4. Experimental layout for single cylinder engine.

Table 2. Measurement facilities for experimental procedure.

equipments name	type	manufacture
pressure censer	4045A	Kistler Ins., Corp.
amplitude	5738	Kistler Ins., Corp.
BNC cables	4761A	Kistler Ins., Corp.
cooling adapter	Z511	Kistler Ins., Corp.
rotary encode	E6C2-CWZ3E	Omron Corp.
A/D board	PCI 6013	National Instruments Corp.
sound level equipment	LM9600	Human Sci.
soft ware	Lab-view6.	National Instruments Corp.

Figure 5. Photograph of experimental model for single cylinder engine.

Figure 4 and Figure 5 show the schematic diagram and photograph of the experimental model for single cylinder engine. The standard type of exhaust pipe was applied as this experimental research. All of the experiments were carried out10 times at every experimental position. Table 2 show the test equipments for experimental model.

The numerical results should be compared with the experimental results, which are required to assess the new engine development and reliability.

4. RESULTS & DISCUSSIONS

Figure 6 shows the experimental results in the case of standard type at d = 22 mm, l = 1,240 mm and N = 1,000 rpm. The pressure curve of P_4 is directly contacted with cylinder, the pressure curve of P_5 is located with middle point of exhaust pipe and the pressure curve of P_6 is placed near ambient. Experimental test timing is ivo to evo. The pressure curve of P_4 shows linear wave after θ = 360°. The pressure curve of P_5 shows exhaust pressure, this is transferred by inner reflection wave and affected by shock wave after θ = 360°. After the first exhaust pressure peak that follows the blow down phase, a second pressure which is due to pressure wave propagation and reflection phenomena in the exhaust system, reaches the exhaust valve during overlap interval, maintaining pressure levels until evc.

Figure 7 shows the numerical results in the case of l = 1,240 mm. The pressure curves of P_4 and P_5 show the transferred exhaust pressure and these show higher values than numerical results, the numerical analysis was assumed that the heat transfer and the friction are not considered in this condition.

Figure 8 shows the comparison between experimental and numerical results in the case of l = 1,240 mm at P_5. The calculation data are quite close to the experimental data, the numerical calculation seems to slightly overestimate over the N = 1,000 rpm, the difference between the computed and experimental data are quite modest.

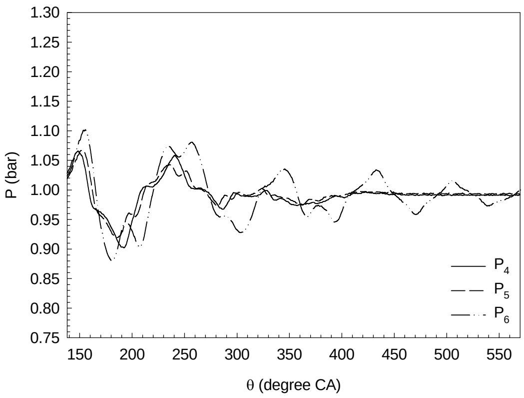

Figure 6. Experimental results in case of standard type at d = 22 mm, l = 1,240 mm and N = 1,000 rpm.

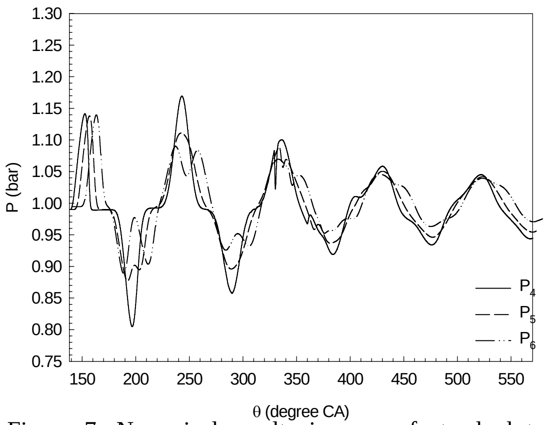

Figure 7. Numerical results in case of standard type at d = 22 mm, l = 1,240 mm and N = 1,000 rpm.

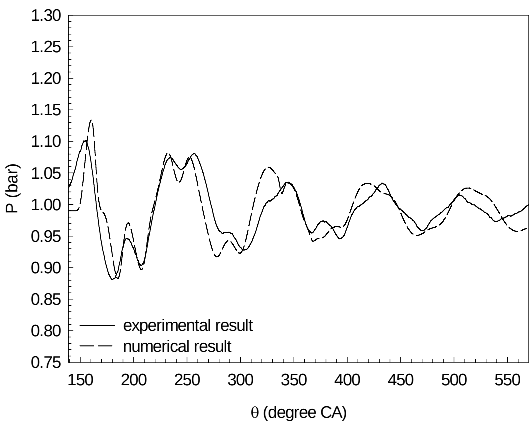

Figure 8. Comparison between experimental and numerical results of pressure distributions for measurement point, P_5 at d = 22 mm, l = 1240 mm and N = 1,000 rpm.

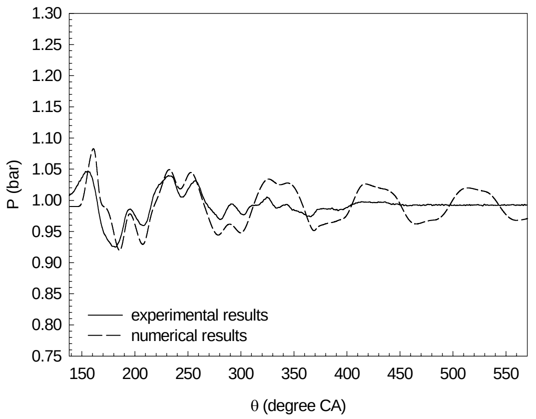

Figure 9. Comparison between experimental and numerical results of pressure distributions for measurement point, P_5 at d = 28 mm, l = 1,240 mm and N = 1,000 rpm.

Figure 9 shows the comparison between experimental and numerical results in the case of d = 28 mm at P5. The calculation data are quite close to the experimental data, the exhaust pressure propagation and reflection phenomena in the exhaust system after θ = 320°, numerical calculation seems to be a slightly overestimation.

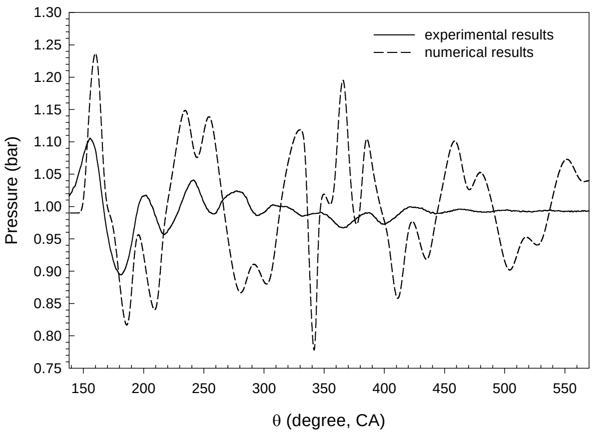

Figure10. Comparison between experimental and numerical results of pressure distributions for measurement point, P_5 at d = 16 mm, l = 1,240 mm and N = 1,000 rpm.

Figure 10 shows the comparison between experimental and numerical results in the case of d = 16 mm at P5. The calculation data are quite different to the experimental data. The reason of this difference is the exhaust pressure propagation, unexpected reflection phenomena and resonant are not consider in the exhaust system, numerical calculation seems to be a slightly overestimation.

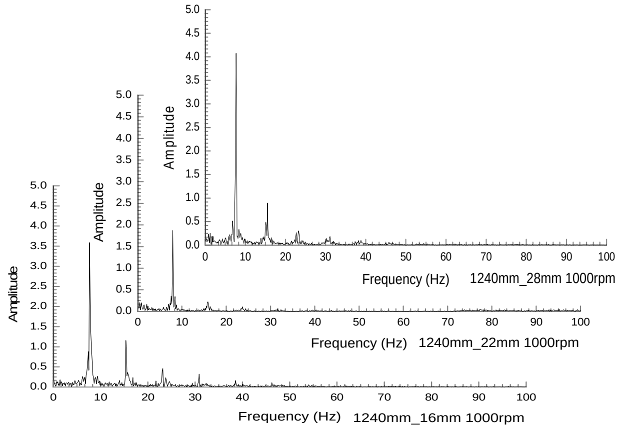

Figure11. Experimental results of Fast Fourier Transform for sound pressure level at d = 16 mm, 22 mm and 28 mm.

Figure 11 shows the experimental results of Fast Fourier Transform for sound pressure level at d = 16 mm, 22 mm and 28 mm. The first frequency of Fast Fourier Transform at N = 1,000 rpm is 8 Hz. In case of d = 22 mm is small amplitude than In case of d = 16 mm and d = 28 mm.

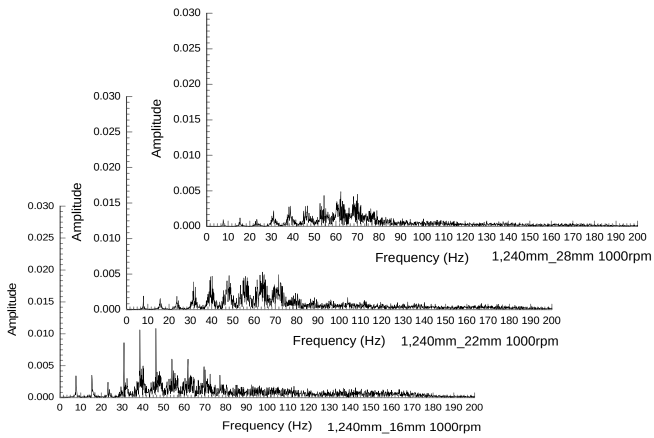

Figure12. Experimental results of Fast Fourier Transform for exhaust pressure at d = 16 mm, 22 mm and 28 mm.

Figure 12 shows the experimental results of Fast Fourier Transform for exhaust pressure at d = 16 mm, 22 mm and 28 mm. The main frequency of Fast Fourier Transform at N = 1,000 rpm is 40 to 80 Hz. In case of d = 22 mm is small amplitude than In case of d = 16 mm and d = 28 mm.

5. CONCLUSIONS

The one dimensional fluid dynamic code has been employed to exhaust pressure, and the predicted data from numerical analysis have been compared with experimental results. Numerical analysis techniques were applied not only to obtain a better understanding of the unsteady flow phenomena that occur during the gas exchange process, but also for the development of a variable geometry exhaust system. In this research, the pressure prediction of exhaust pipe with single cylinder 4-cycle engine is carried out and the main result are as followings. The calculation data are quite close to the experimental data, the numerical calculation seems to slightly overestimate over the various exhaust pipe diameter, the difference between the computed and experimental data are quite modest.

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