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The pulsating pressure in the intake and exhaust manifold of a single cylinder engine by the various of engine revolutions

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Abstract : In this research, a computer analysis has been developed for predicting the pipe pressure of the intake and exhaust manifold in a small single cylinder engine. To get the boundary conditions for a numerical analysis, one dimensional and unsteady gas dynamic calculation is performed by using the MOC(Method Of Characteristics). The main numerical parameters are engine revolutions, to calculate the pulsating flow which the intake and exhaust valves are working. The distributions of the exhaust pipe pressures were influenced strongly to the cylinder pressures, and the shapes of exhaust pressure variation were similar to the inside of cylinder pressure. As the engine revolutions are increased, the intake pressure was lower than ambient pressure. The amplitude of exhaust pressure had increased and the phase of cylinder pressure P_c is delayed and the amplitude of cylinder pressure were increased.

Key words : MOC(Method of Characteristics), Pressure prediction, Intake pipe, Exhaust pipe, Pulsating pressure

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 section area of pipe[m^2]
ivo : intake valve open
ivc : intake valve close
N : engine revolution[rpm]
P : pressure of pipe[bar]
P_c : pressure of cylinder[bar]
tdc : top dead center
U : non-dimensional velocity[u/a_{ref}]
u : gas velocity[m/s]

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β : Rimann variable of characteristics
λ : Rimann variable of characteristics
k : ratio of specific heats, $[C_p/C_v]$
ρ : density$[Kg/m^3]$
θ : crank angle[°]
$\varnothing$: nozzle area ratio$[F_t/F]$
cr : critical conditions
in : characteristic towards boundary
out : characteristic from boundary
t : throat condition

1. Introduction

It is very important to estimate the engine performance before a new engine development. If the development for new engine are required, the manufactor have to change or design with competitive price and low running cost. Thus, the use of computer simulation techniques for the improvement of engine development has been rapidly expanded over the last decade. The development of CFD (Computational Fluid Dynamic) program about pressure prediction of engine manifold is need and so important. The scavenge loss by Mugele at al [1] was simulated to a conventional two-stroke engine by applying a combination of a one dimensional and three dimensional simulation procedures. The automotive emission regulations are increased by the Environmental Protection Agency(EPA), and many of the researches for engine exhaust and manifold are reported. The pressure pulse was reported with various exhaust manifold type by using one dimensional method by Bassett at al [2]. For the pressure prediction of small engine manifold, many numerical calculations are utilized. Generally, the development of CFD program about pressure prediction of pipe was used by method of characteristics [3]. The method of is convenient, because the partial difference equation is converted to total difference equation by Riemann. The various boundary conditions are proposed at engine manifold suggested Benson [4-5].

Researches about exhaust manifold of the automobile engines are increasing, but, these researches are the focused on the pressure prediction of engine cylinder for one dimensional and the unsteady compressible flow dynamics is assumed.

This research for pressure prediction in intake and exhaust manifold mainly focused on the region that the intake and exhaust valves are working. The main calculation parameters are the engine speeds with region from 3,000 rpm to 10,000 rpm.

2. The method of numerical solution and model

2.1 The method of numerical solution

The one dimensional and unsteady compressible fluid dynamics is assumed in this research. The friction and heat transfer were ignored in this research for isentropic flow. The governing equations of continuity, momentum and the characteristics of λ, β are given as followings :

$$\frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial x} + u \frac{\partial \rho}{\partial x} = 0 \quad (1)$$

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

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

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

2.2 Boundary condition

The length of intake and exhaust pipe is measured by real motor cycle that selected for numerical calculation model. The boundary conditions of intake and exhaust manifold for single cylinder 4-stroke engine are explained the followings, and the method of characteristics was introduced for numerical analysis.

Boundary condition of the close end.

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

Boundary condition of the 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.

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

2.3 Numerical model

The specification of model engine as single cylinder 4-stroke engine are shown in Table 1.

Fig. 1 and Fig. 2 show the schematic diagram and photograph of motor cycle engine. The mesh generation is consisted with constant space grid, and the numbers of mesh are sixty. The temperature and ambient pressure of numerical calculation model is 20℃ and 1 bar, respectively.

Table 1 Specifications of a model engine.

engine type	4-stroke, single cylinder
engine displacement (cc)	124.1
cylinder Dia. (mm)	56.5
connecting rod length (mm)	100
stroke (mm)	49.5
intake valve open (°)	348
intake valve close (°)	568
exhaust valve open (°)	138
exhaust valve close (°)	360
compression ratio	9.5

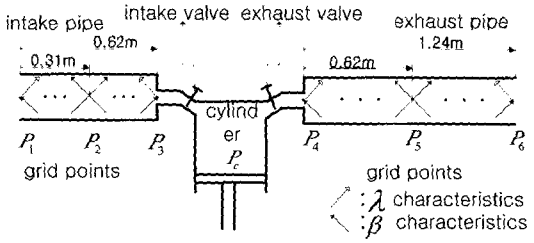

Fig. 1 Schematic diagram of numerical model for single cylinder engine

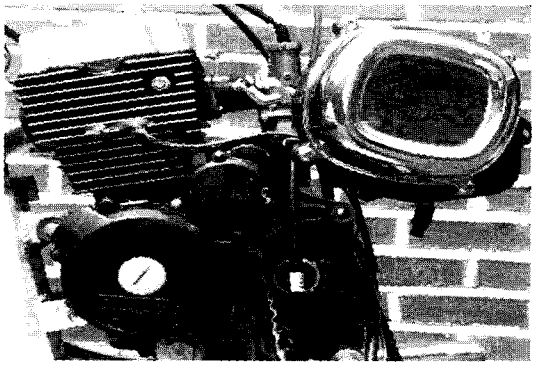

Fig. 2 Photograph of motor cycle engine

Fig. 3 shows the intake and exhaust

valve opening area according to timing. This data was obtained by measurement valve lift data. Intake valve opening timing is 348° and the exhaust valve opening timing is 138°. The intake maximum area is wider with 14 % than the exhaust maximum area.

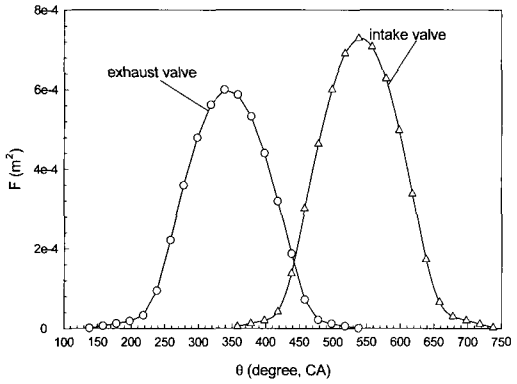

Fig. 3 Intake and exhaust valve opening area according to crank angle

3. Results and discussions

The main numerical calculation parameters are the variation of the engine revolutions to calculate the pulsating flow during the intake and exhaust valve working.

3.1 Pressure distributions at engine speed 3,000 rpm

The measured intake and exhaust pipe lengths are 620 mm and 1240 mm, respectively.

Fig. 4 shows the cylinder pressure at N=3,000 rpm. The intake pipe is directly connected to the cylinder. The cylinder pressure is dropped until θ =350°. The cylinder pressure was arisen during θ =350° to θ =450°. When the cylinder pressure has been arisen again, for instance, θ =550° and θ =650°. The reason of this pressure drop affected with intake valve open.

Fig. 5 shows the intake pressure variations. The pressure P_1 is directly contacted to ambient, so the pressure value appears similar to ambient pressure. The pressure P_3 is located near to cylinder, which controlled inlet air amount by valve lift. The minimum value of intake pressure is 0.86 bar at θ =550°. The shape of P_3 in Fig. 5 after θ =360° crank angle is similar to the shape of P_c at Fig. 4.

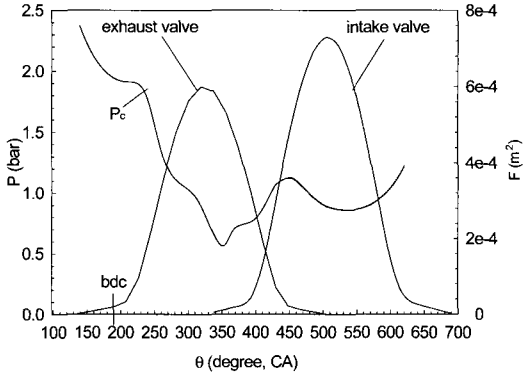

Fig. 4 Cylinder pressure at N=3,000 rpm

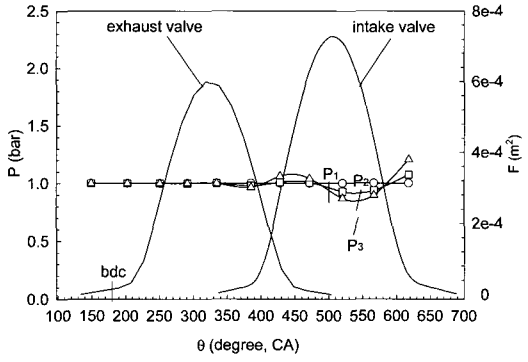

Fig. 5 Intake pressure at N=3,000 rpm

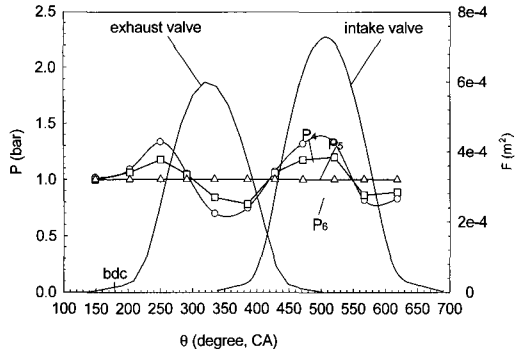

Fig. 6 Exhaust pressure at N=3,000 rpm

Fig. 6 shows the exhaust pressure variation. The pressure P_6 is similar to ambient pressure and have a strait line with value of 1 bar. As the pressure distribution of P_4 is directly contacted with cylinder, the shape of pressure P_4 is similar to the cylinder pressure after exhaust valve opening timing. When the crank angle is increased, the first peak of minimum value at pressure P_4 is 1.34 bar and the peak pressure P_5 is 5 % lower than the peak pressure P_4.

3.2 Pressure distributions at engine speed 6,000 rpm

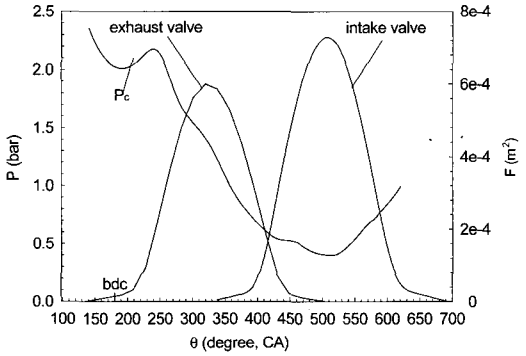

Fig. 7 Cylinder pressure at N=6,000 rpm

Fig. 7 shows the cylinder pressure variation at N=6,000 rpm. The cylinder pressure has dropped under the ambient pressure until $\theta = 520°$. The minimum cylinder pressure is 0.4 bar at $\theta = 500°$, and the minimum value of cylinder pressure at N=6,000 rpm is 0.2bar lower than at N=3,000 rpm.

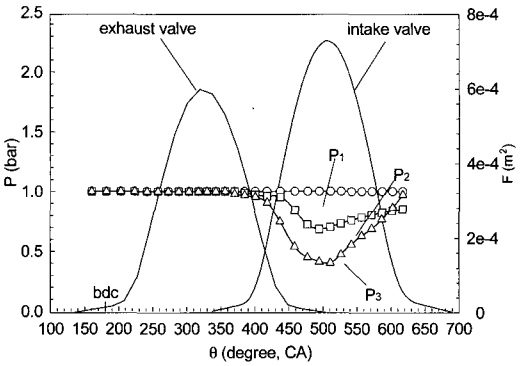

Fig. 8 Intake pressure at N=6,000 rpm

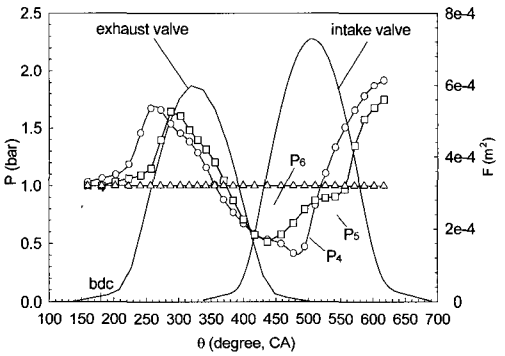

Fig. 9 Exhaust pressure at N=6,000 rpm

Fig. 8 shows the intake pressure variation at N=6,000 rpm. The shapes of intake pressure P_1, P_2 and P_3 at Fig. 8 are similar to shape of cylinder pressure P_c after intake valve opening(ivo) timing. The lowest intake pressure is 0.4 bar at $\theta = 500°$, the lowest value at N=6,000 rpm is 0.46 bar lower than N=3,000 rpm. The increasing engine revolution cause a rise of the pressure drop. The shape of intake pressure directly affected to the

shape of cylinder pressure after intake valve opening timing. The difference of pressure P_2 and P_3 is 0.1 bar.

Fig. 9 shows the exhaust pressure variation at N=6,000 rpm. The shape of P_6 is not changed, because the pressure P_6 directly contacted with ambient, and that pressure value is similar to ambient pressure. The amplitude of pressure wave at the point P_4 is 0.2 bar higher than one at the point P_4 in Fig. 7. The first peak of maximum value is 1.69 bar at θ =270° in the pressure P_4, which is 0.35 bar higher than the pressure P_4 in Fig. 6. The peak pressure P_5 is almost same value of the peak pressure P_4.

3.3 Pressure distributions at engine speed 10,000 rpm

Fig. 10 shows the cylinder pressure variation at N=10,000rpm. The cylinder pressure has dropped under the ambient pressure until θ =540°. The shape of cylinder pressure has been changed more simplicity. The minimum cylinder pressure is 0.4 bar at θ =540°, the minimum value of cylinder pressure at N=6,000 rpm is 0.2 bar lower than one at N=3,000 rpm.

Fig. 11 shows the intake pressure variation at N=10,000 rpm. The shapes of intake pressure P_1, P_2 and P_3 at Fig. 11 are similar to one of cylinder pressure P_c after intake valve opening(ivo) timing. The lowest intake pressure is 0.44 bar at θ =530°, the lowest value of N=10,000 rpm is 0.42 bar lower than N=3,000 rpm. The higher engine speed caused the improvement of the pressure drop at both intake pipe and cylinder. The shape of intake pressure directly affected to the shape of cylinder pressure after intake valve opening point. The difference of pressure P_2 and P_3 are 0.21bar. The higher engine speed caused the increasing the pressure difference between intake pressure P_2 and P_3.

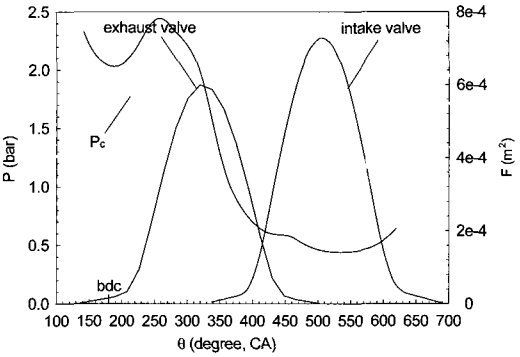

Fig. 10 Cylinder pressure at N=6,000 rpm

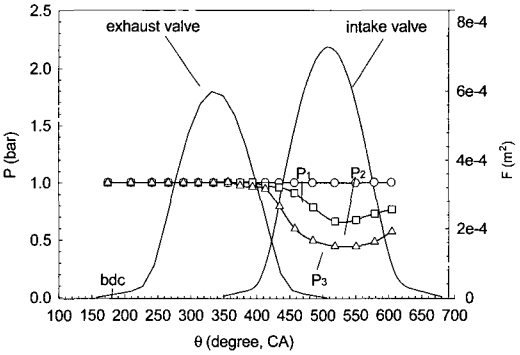

Fig. 11 Intake pressure at N=10,000 rpm

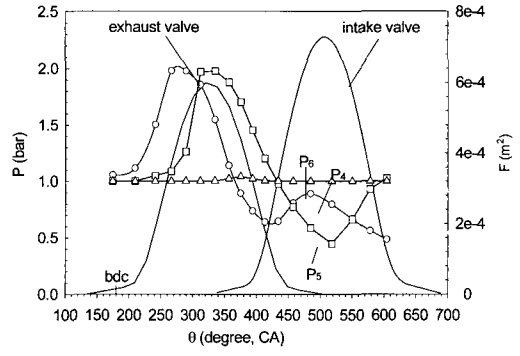

Fig. 12 Exhaust pressure at N=10,000 rpm

Fig. 12 shows the exhaust pressure variation at N=10,000 rpm. The shape of P_6 is not changed because the pressure P_6 directly contacts with ambient, that pressure value is similar to ambient pressure. The amplitude of pressure wave at the point P_4 is 0.4bar higher than the point P_4 in Fig. 6. The first peak of maximum value is 2.03 bar, at θ =275°, of pressure P_4, which is 0.72 bar higher than the pressure P_4 in Fig. 5. Also, the wave phase was delayed about 25° of crank angle. Raising the engine speeds caused the increasing amplitude of exhaust pressure to increase. The peak pressure P_5 is almost same value of the peak pressure P_4.

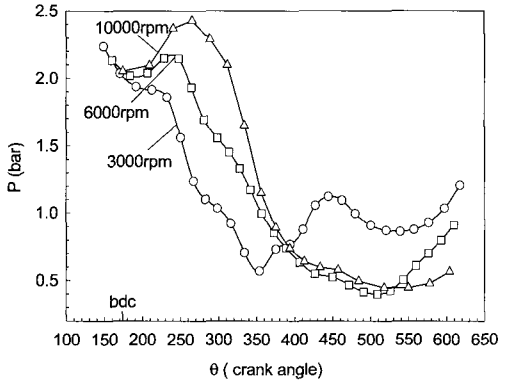

Fig. 13 The various cylinder pressures versus crank angle.

Fig. 13 shows the various cylinder pressure versus crank angle for the amplitude of cylinder pressure at each engine speed: 3,000 rpm, 6,000 rpm and 10,000 rpm, respectively. When the engine revolution is increasing, the amplitude has been arisen by the acceleration of piston speed near the region under the crank angle θ =300°. From the comparison between the 3,000 rpm and 6,000 rpm, the phase of the minimum cylinder pressure P_c is delayed about 150°. Also, the phase of the minimum cylinder pressure in case of 10,000 rpm is delayed about 30° in case of 6,000 rpm.

5. Conclusion

The results of this research about pressure prediction of intake and exhaust manifold with single cylinder 4-stroke engine are similized as followings.

(1) As the engine revolution is increased, the intake pressures, P_2 and P_3, become lower after remaining at ambient pressure until crank angle θ =400°.

(2) As the engine revolution is increased, the pulsating amplitude of exhaust pressure had increased.

(3) The first peak of cylinder pressure was showed in case of 6,000 rpm at θ =250°, this peak is delayed about 30° in case of 10,000 rpm.

(4) As the engine revolutions are increased, the phase of cylinder pressure P_c is delayed and the pulsating amplitude of cylinder pressure were increased.

Acknowledgements

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