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AUTOMOTIVE ENGINE SOUND CREATION BY SPECIFYING A SMALL NUMBER OF PARAMETERS

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ABSTRACT

This paper shows a method of synthesizing engine combustion sound by a computer for evaluating engine design at an early stage of its development. The method used was a kind of physical modeling of engine combustion sound generation. The excitation gas force was modeled by a Wiebe's heat release function. The transmission routes were represented by modal functions that were obtained through finite element modeling or impact excitation measurement. In this modal experiment, the transfer functions between the combustion pressure and the engine structural vibration were analyzed making use of Maxwell's reciprocal theorem. Through subjective evaluation of the synthesized sound, fluctuation among the vibration responses caused by the combustion pressure at different cylinders and cycles was the key for the sound to be perceived as realistic engine sound.

1 - INTRODUCTION

Sound quality of a car is an important factor that represents the car's overall character and quality. The sound quality is determined by the design architectures of the car, and cannot be improved so much by detail design at a later design stage. The design architectures in this case are, for instance, fundamental structural dimensions of the engine and combustion process. Therefore, it is highly desirable for a designer to evaluate the car design in terms of its sound quality at an early stage of development.

1.1 - Requirements for the sound creation as a tool of engine design evaluation

- Simple modeling: At an early stage of design, precise information of the product is not fixed. The design factors sometimes remain in conceptual phase.
- Simple method but still being capable to represent the physical meaning: In this case, such design factors as combustion process and engine structure vibration mode parameters were considered.
- Simple software and simple hardware: Designers should be able to evaluate the sound by themselves in an office. Running a structural analysis software on a super computer is too complicated for the above-mentioned usage.

1.2 - Personal computer technology as an answer to the requirements

If we look at usable signal-processing devices, personal computer hardware and software have grown to have enough multimedia functions that can process complex signals by simple command lines or can generate physical sound wave from numerical data. [1]-[2] In this study, automotive engine sounds are synthesized using so-called physical modeling method on a personal computer by specifying small number of design parameters. The sounds are evaluated subjectively to understand combustion speed effect and vibration characteristics effect on sound quality.

2 - METHODS OF SYNTHESIZING SOUND

2.1 - Overview of available technology

There are five typical methods in synthesizing sound by computers. They are frequency modulation method, subtraction method from recorded data, analytic method using special signal generating functions, synthesizing filtered signals method, and physical modeling method. The first three are well-established methods, but they do not have direct relationship with product design. Synthesizing filtered signals has physical meaning to some extent such as creating vowels. However, the last one is the only method that can be cooperatively used with design. In this method, the excitation force models and the vibration or acoustic power transfer path models must be programmed in a computer. In this study, the excitation forces were combustion pressure inside each cylinder of four-cylinder four-stroke engine.

2.2 - Excitation force modeling

In order to simulate the cylinder pressure waveform as close to real one as possible, the cycle was represented by the three phases as shown in the figure 1, i.e. adiabatic compression, combustion process and adiabatic expansion intervals. The intake and exhaust stroke were treated as zero pressure intervals.

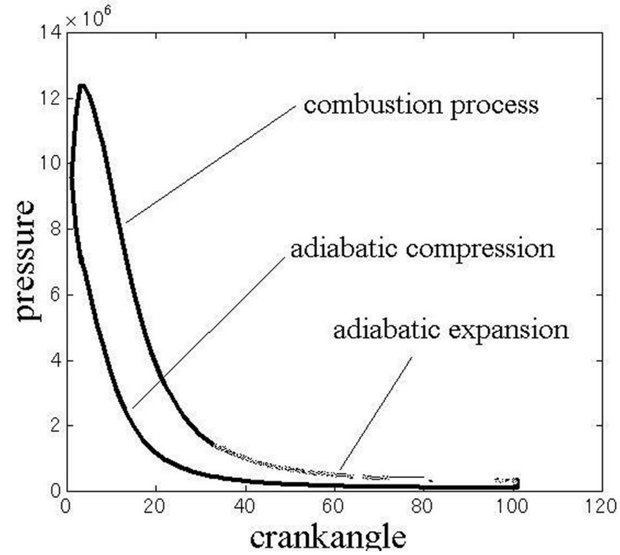

Figure 1: The cylinder pressure cycle on P-V diagram.

The combustion process was described as shown in the figure 2. The first interval is pre-combustion process and expressed by a triangular waveform. The second interval is diffusive combustion process and expressed by Wiebe's heat release function. The equations used to calculate the heat release rate are described as below.

1) The first half of the pre-combustion: $\theta_1 \leq \theta_i \leq \theta_2$

Here, θ_1, θ_2, θ_3 and θ_4 are the crank angles corresponding to the positions marked from 1 to 4 in the figure 2. The θ_i is the arbitrary crank angle at which the heat release rate is calculated.

Let $\varphi = \theta_i - \theta_1$, then

$$\frac{dQ}{d\varphi} = \frac{\left(\frac{dQ}{d\theta}\right)_{\theta_2}}{d\theta} \tag{1}$$

Where $\left(\dfrac{dQ}{d\theta}\right)_{\theta_2}$ is a constant with the dimension of kcal/deg.

By integrating the equation (1), the overall heat supplied to the combustion gas from the onset of combustion to θ_i can be expressed as the next equation.

$$Q_i = \int \frac{\left(\frac{dQ}{d\theta}\right)_{\theta_2}}{d\theta}\varphi d\varphi = \frac{\left(\frac{dQ}{d\theta}\right)_{\theta_2}}{d\theta}\varphi^2 \qquad \text{[kcal]} \tag{2}$$

2) The second half of the pre-combustion: $\theta_2 \leq \theta_i \leq \theta_3$

Now, let $\varphi_3 = \theta_3 - \theta_i$, and $\varphi_0 = \theta_4 - \theta_i$. Substituting the crank angle φ by $\varphi = \varphi_3$ in the Wiebe's heat release function, we get the next equation.

$$\left(\frac{dQ}{d\theta}\right)_{\theta_3} = \left(\frac{dQ}{d\varphi}\right)_{\varphi_3} = \frac{Q_{\ 0}}{\varphi_0}a\,(m{+}1)\left(\frac{\varphi_3}{\varphi_0}\right)\exp\left\{-a\left(\frac{\varphi}{\varphi_0}\right)^{(m+1)}\right\} \tag{3}$$

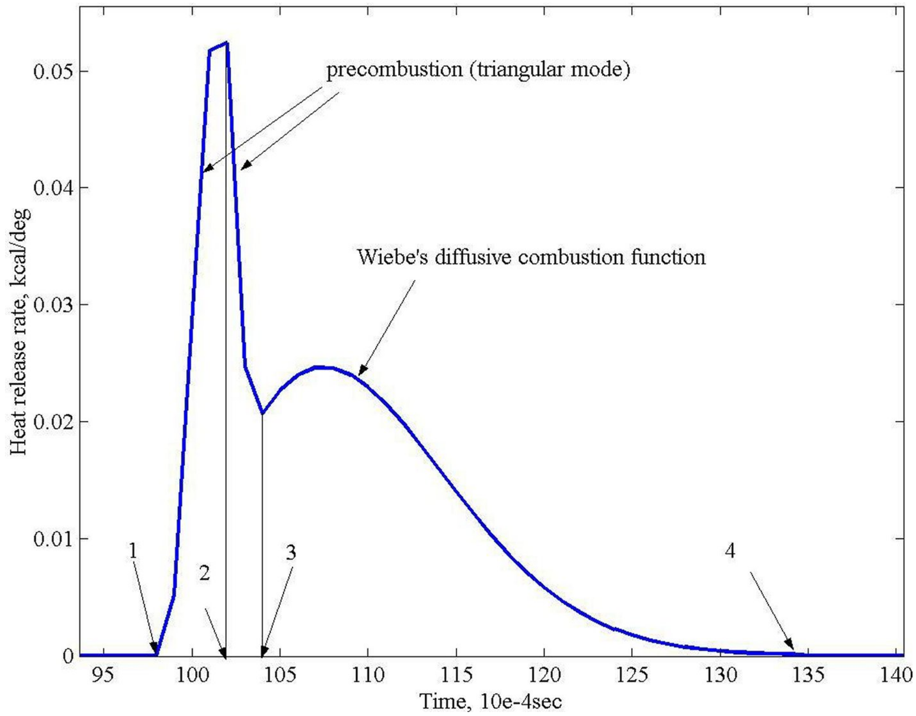

Figure 2: Heat release rate diagram.

The constant m in this equation changes the location of maximum combustion along the crank angle coordinate. The equation (3) determines the heat release rate at the junction of the pre-combustion and the diffusive combustion. The second half of the pre-combustion heat release curve can be determined as the line connecting the top of the pre-combustion and the Wiebe's diffusive combustion onset as below.

$$\left(\frac{dQ}{d\theta}\right) = \left(\frac{dQ}{d\varphi}\right) = \left\{\frac{\left(\frac{dQ}{d\theta}\right)_{\theta_3} - \left(\frac{dQ}{d\theta}\right)_{\theta_2}}{d\theta}\right\} + \left(\frac{dQ}{d\theta}\right)_{\theta_2} \tag{4}$$

The overall heat Q_i supplied to the gas from the beginning of the second half of the pre-combustion to θ_i can be expressed as the next equation.

$$Q_i = \left\{\frac{\left(\frac{dQ}{d\theta}\right)_{\theta_3} - \left(\frac{dQ}{d\theta}\right)_{\theta_2}}{d\theta}\right\}\varphi^2 + \left(\frac{dQ}{d\theta}\right)_{\theta_2}\varphi + Q_{\theta_2} \tag{5}$$

3) Diffusive combustion interval: $\theta_3 \leq \theta_i \leq \theta_4$
The heat release rate in this interval was represented by Wiebe's heat release function like the equation (3). The overall heat released in this interval can be calculated from the equation (6),

$$Q_i = Q_{wi} - Q_{w3} + Q_{\theta_3} = Q_0\left[\exp\left\{-a\left(\frac{\varphi_3}{\varphi_0}\right)^{(m+1)}\right\} - \exp\left\{-a\left(\frac{\varphi}{\varphi_0}\right)^{(m+1)}\right\}\right] + Q_{\theta 3} \tag{6}$$

where $\varphi = \theta_i - \theta_1$, and Q_{wi} is the heat that would be released up to the crank angle φ when the pre-combustion triangular function would not be used.
4) Cylinder pressure calculation using heat release functions and piston movement.
Considering the thermal capacity of the gas in the cylinder and the work done by adiabatic expansion, the cylinder pressure P_i can be obtained as the equation (7).

$$P_i = P_{i-1} + dP_i = \frac{P_{i-1} + \frac{k_i - 1}{AV_i}dQ_i}{AV_i} \quad \text{[Pa]} \tag{7}$$

The nomenclatures in the equation (7) are as follows:

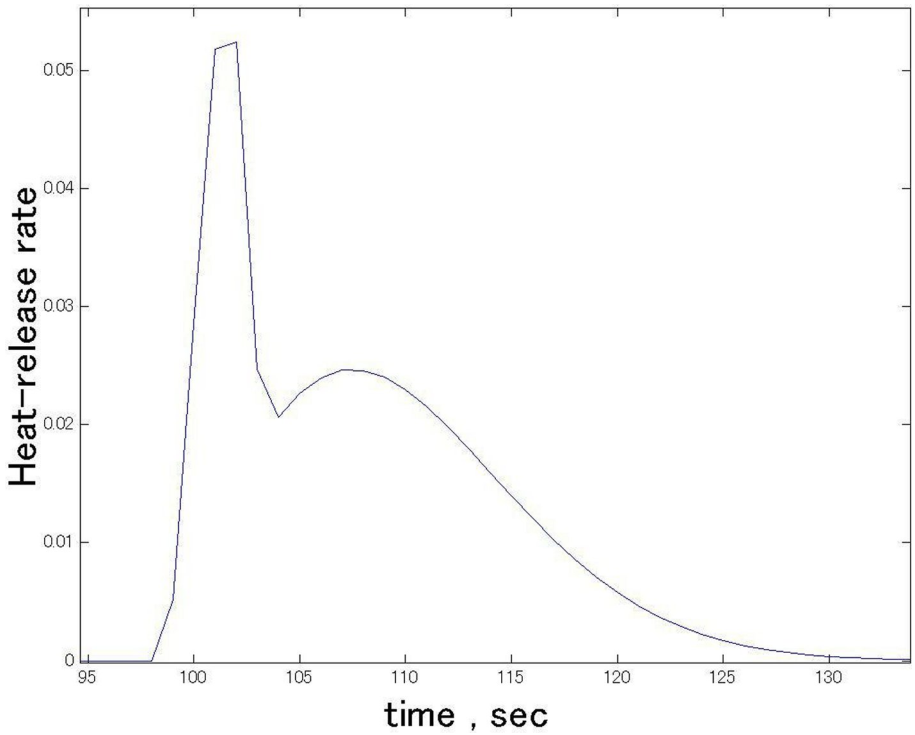

Figure 3: Heat release rate diagram.

- k_i: specific heat ratio at the crank angle θ_i,
- A: mechanical equivalent of heat,
- V_i: combustion chamber volume at the crank angle θ_i.

$$V_i = V_0 + rS\left\{1 - \cos\theta_i + \frac{l}{r} - \sqrt{\left(\frac{l}{r}\right)^2 - \sin^2\theta_i}\right\} \qquad [\mathrm{m}^3] \tag{8}$$

where, l: the connecting rod length, r: the crank arm length, S: piston top surface area.
A typical cylinder pressure waveform calculated is shown in the figure 4. The numerical values of indices used in this calculation are set at the values as follows.
θ_1 =-4, θ_2 =1, θ_3 =3 and θ_4 =60 in degrees crank angle, a=7.637, b=0.953, Q_0=0.66 [kJ], l=0.225 [m], r=0.075 [m], piston diameter=135 [mm] A=0.0239e^{-3} [kcal/Nm], ki=1.35.

2.3 - Structural vibration characteristics modeling

For the simplicity, modal expression was adopted for the modeling of the engine structural vibration characteristics. Major sound quality problems of automotive engine that are caused by the fundamental structural design appear in the frequency range from 250 Hz to 1000 Hz. It is common to have several major elastic vibration modes in this range. The modes are not simple. However, torsion and lateral bending of the cylinder block and the relatively flexible joining in between the engine and the transmission usually dominates the deformation shapes.
Because each vibration mode is independent each other, the overall response is the summation of the each mode response. If we use time domain expression, then the structural modeling for the sound creation can be described as in the equation (9).

$$\sum_{i=1}^{N} h_i\left(t\right) = \sum_{i=1}^{N} \frac{1/m_i}{\omega_{di}} \exp\left(-\zeta\omega_{0i}t\right)\sin\left(\omega_{di}t\right) \tag{9}$$

Here, the $h_i(t)$ is the impulse response function of the ith mode, and the terms used are

$$\omega_{di} = \omega_{0i}\sqrt{1-\zeta_i^2}, \qquad \zeta_i = \frac{c_i}{2\sqrt{m_i k_i}}, \qquad \omega_0 = \sqrt{\frac{k_i}{m_i}}$$

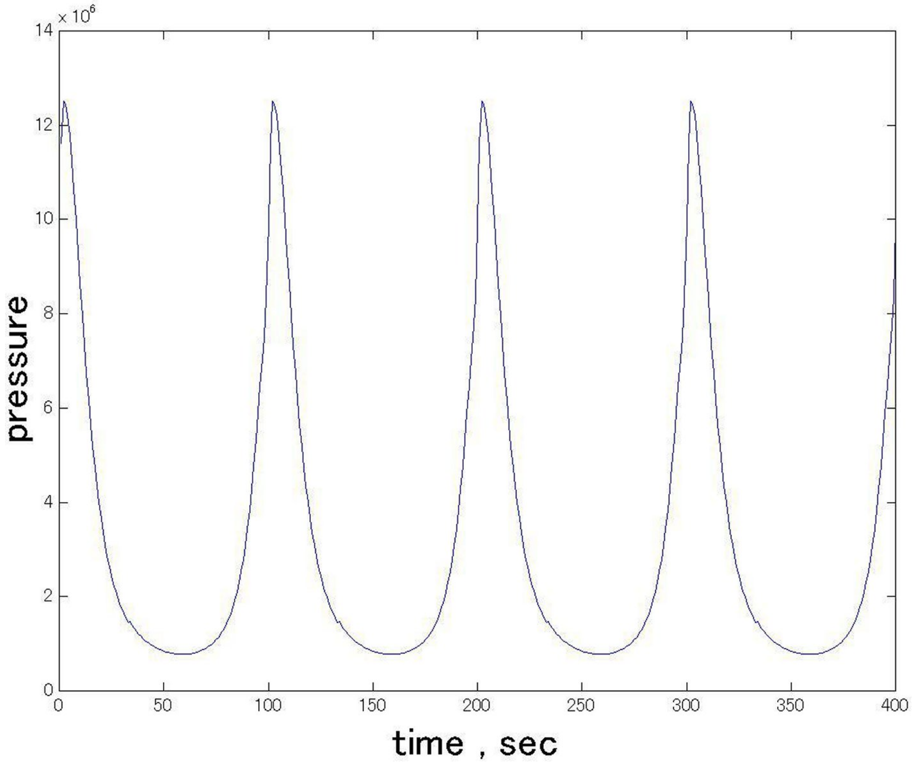

Figure 4: Cylinder pressure waveform.

The m_i, c_i and k_i are the equivalent modal mass, damping and stiffness of the ith mode.
By using this modal modeling, only a small number of parameters are required to represent the vibration response. This is a very important advantage in the design revue at an early development stage. In this study, the impulse response functions were obtained by experimental modal analyses. An important part of this analysis was the measurements of the cylinder pressure as responses to the vibration input by an impact hammer. This method was adopted because the transfer function from cylinder pressure to the vibration output was not be obtained due to the overlapping effect under usual engine operating conditions.
Typical transfer function and impulse response function used in this study are shown in the figures (5) and (6). The data used were as follows. k_1=8.12 $\times 10^6$, k_2=1.41 $\times 10^6$, k_3=1.41 $\times 10^6$ [N/m], c_1=750, $c_{\ 2}$=350, c_3=350 [Ns/m], m_1=100, m_2=10, m_3=7 [kg].

2.4 - Sound synthesis using Matlab

The sound waveforms $y(t)$ were calculated by the convolution integral of the cylinder pressure and impulse response functions as in the next equation.

$$y\left(t\right)=\int_{-\infty}^{t}h\left(\tau\right)P\left(t-\tau\right)d\tau \tag{10}$$

Then, this sound waveform data were transferred to wav format in the personal computer and fed to audio headphones. The data are shown in figures 7, 8 and 9.

3 - SUBJECTIVE EVALUATION OF THE SYNTHESIZED SOUND

Though the above-mentioned method seemed to be a physically reasonable one, sound using periodical gas force excitation and the same transfer functions for all cylinders generated electric motor like sound rather than engine sound at a constant engine speed. On the other hand, the sounds were accepted as real engine sound under accelerating conditions.
Synthesized sound using periodical gas force excitation and different transfer functions for all cylinders obtained more favorable evaluation. Sound generation using fluctuated excitation and different transfer functions for each cylinder produced the best results though it did not get full scores from the panelists. A number of design factors were examined subjectively. The sound synthesizing system that was developed in this system turned out to very easy to use. The most influential design factor was damping

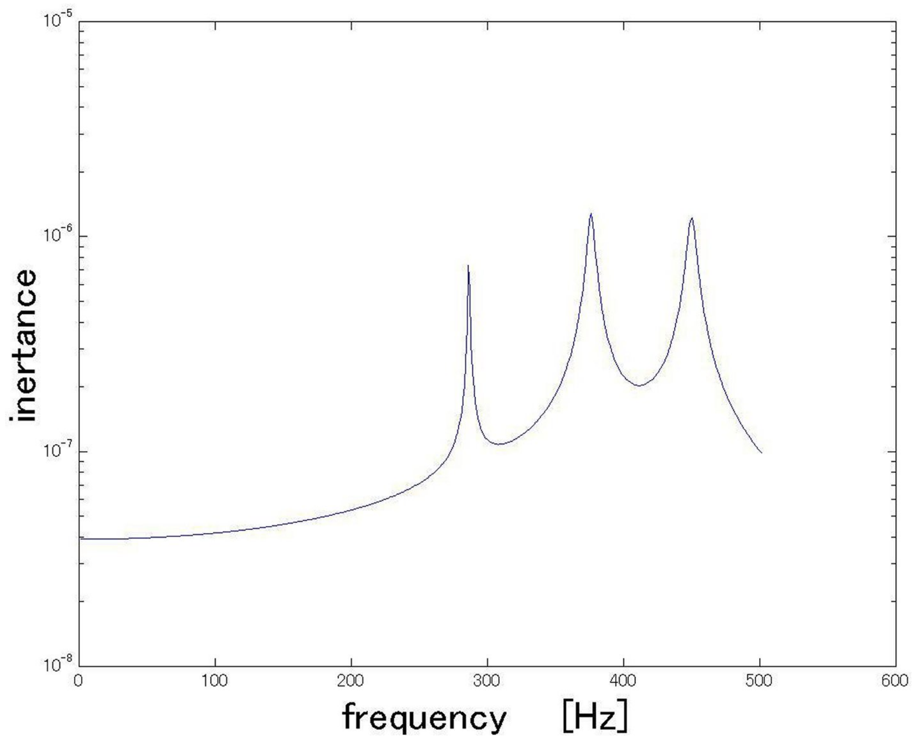

Figure 5: Typical transfer function with three modes.

ratio, ζ. However, we did not examine combination effect of the factors.

4 - CONCLUSIONS

- A new physical modeling method was developed to synthesize automotive engine sound.
- This method can produce real-engine-like sound by specifying small number of design parameters.
- Influence of some design factors were evaluated by listening to the sound synthesized.

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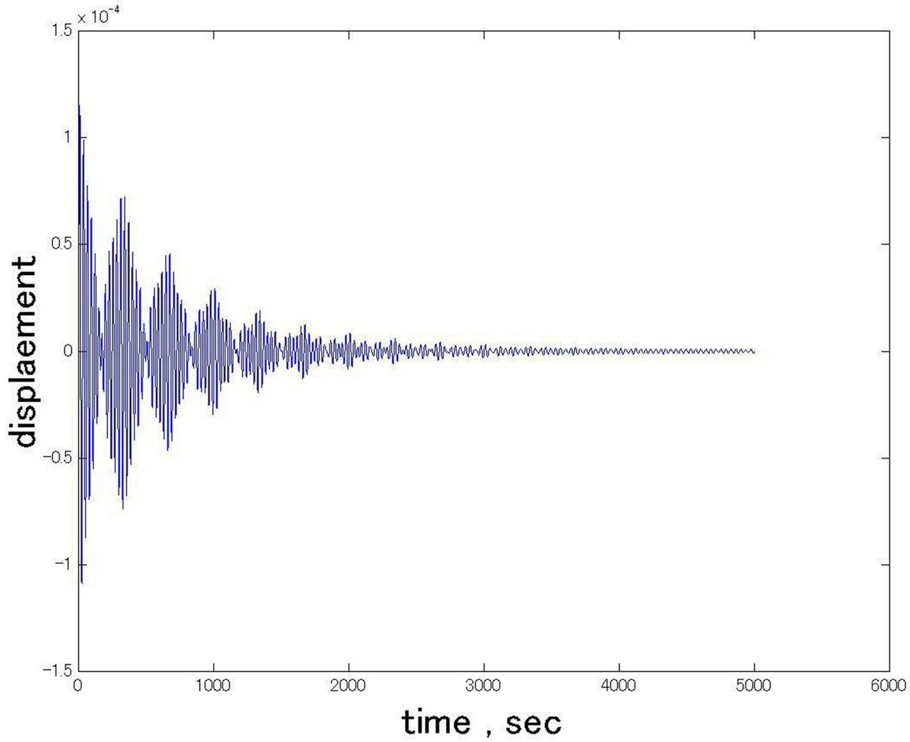

Figure 6: Typical impulse response function.

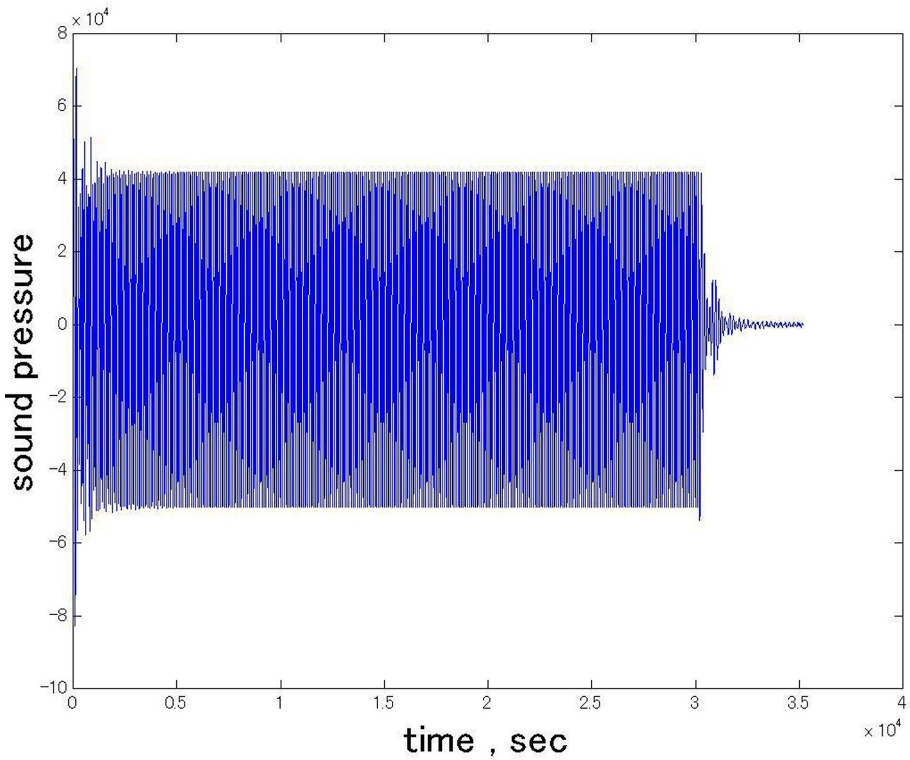

Figure 7: Sound waveform at 3000 rpm.

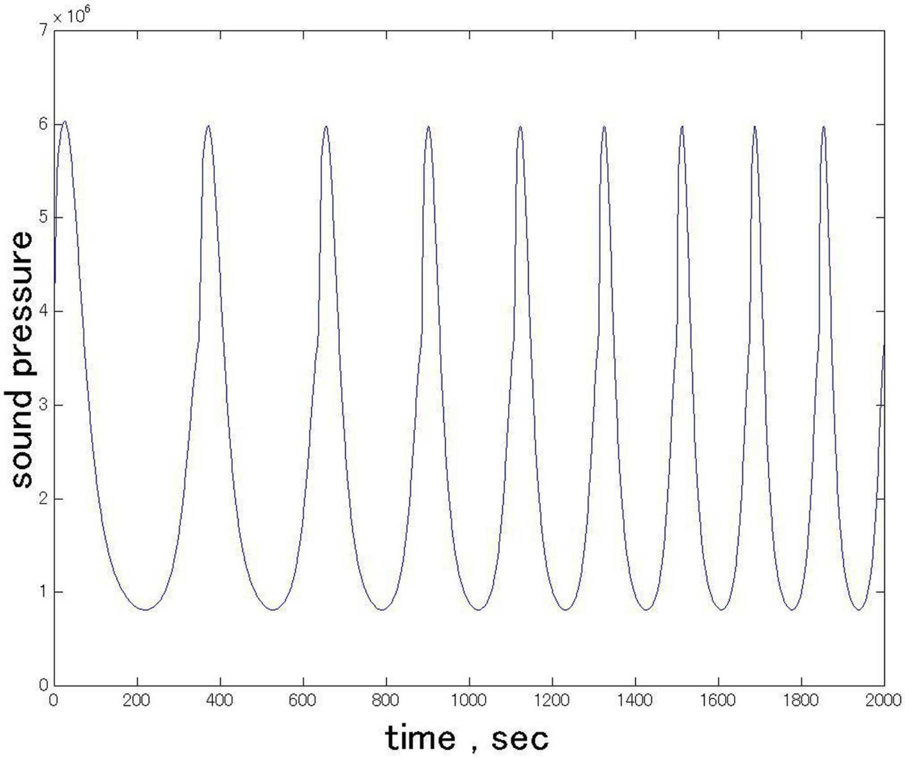

Figure 8: Cylinder pressure during acceleration.

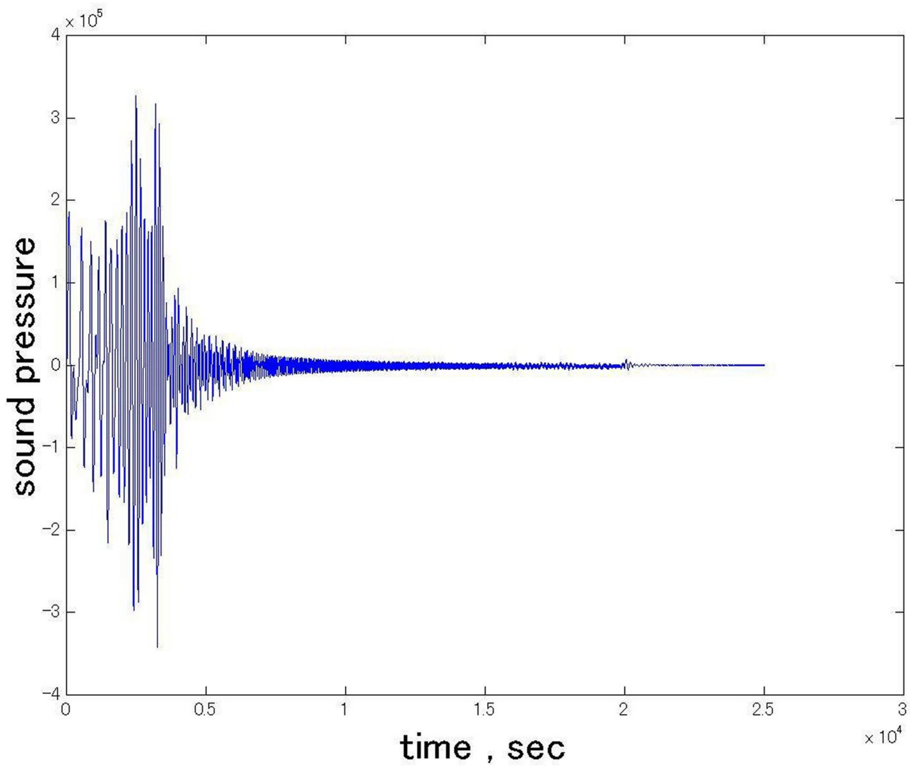

Figure 9: Sound waveform during acceleration.