Analysis of Piston Slap Induced Noise and Vibration of Internal Combustion Engine
(Effect of Piston Profile and Pin Offset)*

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Abstract
This paper presents the analytical method to evaluate the impact force and engine noise induced by piston slap considering the detailed piston profile and dynamic characteristics of the piston. This method was applied to examine the effect of the piston profile and piston pin offset on the piston slap impact force and its induced engine noise. Present numerical method showed that increasing the upper clearance of the piston skirt enlarged the higher frequency components of the piston impact force and engine noise and this was verified by the experimental results. The analysis revealed that piston movement at the vicinity of the combustion top dead center was governed by the friction moment around the piston pin and the additional piston pin offset moment induced by combustion pressure could cancel this friction moment and reduce the piston impact force if the piston pin was offset to the major thrust side. This analytical results was confirmed by the measured results of actual diesel engine.

Key words: Piston Slap, Internal Combustion Engine, Impact Vibration, Noise Control, Modal Analysis

1. Introduction

Many intensive researches on the combustion control of engine have been conducted to meet the stringent emission regulation using advanced electric control technology such as high pressure and multi-staged fuel injection or adjusting ignition timing. On the other hand, it is required not only to reduce the level of noise and vibration of engine that give the discomfort and fatigue to the operator or people around it but also to improve sound quality of engine suitable for working and living environment. One of the major factors that affect the engine sound quality is the piston slap in which the piston collides with the liner at high speed and it has a large influence on the high frequency components of the engine noise spectrum. One of the authors (1), (2) has developed the analytical method to predict the piston slap force and its induced dynamic response of engine components like engine block. Recently, Koizumi (3), (4) presented the numerical procedure to predict piston slap considering the detailed piston profile and oil film of lubricant. Author presented the piston impact model of two collision points on each side in the reference (1) and modified multipoint
Collision model is offered in this paper to discuss the effect of detailed piston and liner profiles on the piston motion and impact force. Furthermore, validity of piston pin offset to reduce the piston slap and engine noise is examined by numerical analysis and measurement on actual diesel engine.

**Nomenclature**

- \( \alpha \): Rotational angle of crankshaft
- \( \beta \): Tilting angle of the connecting rod
- \( \gamma_0 \): Angle between the vertical axis and the line connecting the crankshaft center and piston pin at the top dead center
- \( \mu_p \): Friction coefficient between piston pin and bearing
- \( D \): Mean diameter of piston
- \( F_{Ai} \): Distributed impact force between piston and liner on the thrust side
- \( F_{Ei} \): Distributed impact force between piston and liner on the anti-thrust side
- \( F_f^\alpha \): Friction force in vertical direction between piston skirt and liner
- \( F_g \): Gas force
- \( F_{IX} \): Inertia force due to piston motion in transverse direction
- \( F_{IV} \): Inertia force due to piston motion in vertical direction
- \( F_{Kr} \): Restoring force of corrugated piston ring in transverse direction
- \( F_l \): Axial force of connecting rod
- \( F_{qj} \): Friction force between the j-th piston ring and the liner in vertical direction (\( j=1\sim3 \))
- \( F_{rf} \): Friction force between the j-th piston ring and the ring groove in transverse direction (\( j=1\sim3 \))
- \( H_i \): Height of point i from the center of piston pin
- \( I_G \): Moment of inertia around the center of gravity of piston
- \( m_p \): Mass of piston
- \( m_r \): Mass of piston ring
- \( m_{rg} \): Sum of mass of piston pin and equivalent mass of connecting rod's small end
- \( l_0 \): Distance between piston pin at the top dead center and crankshaft center
- \( l \): Length of connecting rod
- \( r \): Crank radius
- \( T_I \): Moment due to piston's rotational motion around piston pin
- \( T_P \): Moment around piston pin induced by friction force
- \( x_{ca} \): Offset of crankshaft
- \( x_{po} \): Offset of piston pin
- \( (x_G, y_G) \): Coordinate of piston gravity center
- \( (x_P, y_P) \): Coordinates of piston pin

**2. Analysis of piston slap and engine noise prediction**

**2.1 Numerical procedure to predict piston slap**

Fig.1 shows the coordinate system in which y axis is set on the centerline of the cylinder liner and upward is positive. At the top dead center, x axis is set on the piston pin center and is perpendicular to y axis and its positive direction is taken to the anti-thrust side of liner. Rotational angle of piston \( \theta_p \) is positive in counterclockwise. Piston profiles of thrust side and anti-thrust side after thermal deformation are approximated by sequence of points.

**1) Equation of motion of piston**

As the piston collides with the liner while moving, the displacement of the piston is expressed by linear combination of rigid body motion and elastic vibration. Rigid body motion of piston is represented by the transverse displacement of piston pin \( x_P \) and
Rotational angle of $\theta_p$ around piston pin. Equation of piston's rigid body motion is given by the matrix form as follows\(^{(2)}\)

$$
\begin{bmatrix}
m_p + m_r & -m_p (L_y - L_x \tan \beta) \\
-m_p L_y & I_p
\end{bmatrix}
\begin{bmatrix}
\dot{x}_p \\
\dot{\theta}_p
\end{bmatrix}
= \begin{bmatrix}
F_{xp} \\
T_{\theta p}
\end{bmatrix}
$$

(1)

where $L_x = x_g - x_p$, $L_y = y_g - y_p$ and force $F_{xp}$, moment $T_{\theta p}$ acting on piston pin are expressed by

$$
F_{xp} = -F_i \sin \beta + \sum_i F_{Al} - \sum_i F_{El} + \sum_j F_{ij} + F_{kr}
$$

$$
T_{\theta p} = -m_p L_y \dot{y}_p + T_p + F_{xp} x_{p0} - \sum_i F_{Al} \cdot H_i + \sum_i F_{El} \cdot H_i
$$

$$
- F_{f1} \left( \frac{\dot{I}_y}{2} + x_{p0} \right) + F_{f2} \left( \frac{\dot{I}_y}{2} - x_{p0} \right) + \sum_j F_{ij} \cdot H_{ij} + F_{kr} \cdot H_{kr}
$$

Friction coefficient between piston pin and bearing is defined as $\mu_p$ and radius of piston is $R_p$. Friction moment $T_p$ between piston pin and bearing is caused by the reaction force of connecting rod $F_i$ and is given by the following equation by use of rotational velocity of piston $\dot{\theta}_p$ and tilting angular velocity of connecting rod $\beta$

$$
T_p = \text{sgn}(\phi) \mu_p R_p F_i
$$

(2)

where

$$
\phi = \dot{\theta} - \dot{\theta}_p , \quad \text{sgn}(\phi) = \begin{cases} 
1 & (\phi > 0) \\
-1 & (\phi \leq 0)
\end{cases}
$$

Impact force at point i of piston is expressed by the following equation by use of contact spring coefficient $k_{Ai}$ and oil film damping coefficient $c_{Ai}$

$$
F_{Ai} = -k_{Ai} (x_{PAi} - x_{LAi}) - c_{Ai} (\dot{x}_{PAi} - \dot{x}_{LAi})
$$

(3)

Fig. 1 Analytical model of piston slap
Elastic vibration amplitude of piston is given by the superposition of eigen mode $\phi_n$.

$$x_{pi} = \sum_{n=1}^{N_{pi}} \phi_{i,n} \cdot a_n(t) \quad (4)$$

The equation of motion of the n-th mode of piston $a_n$ is

$$\ddot{M}_{p,n} \dot{a}_n + 2\zeta_n \omega_n \dot{M}_{p,n} a_n + \ddot{M}_{p,n} \omega_n^2 a_n = \ddot{f}_{p,n} \quad (5)$$

where $\ddot{M}_{p,n}$ is the modal mass, $\zeta_n$ is the damping ratio and $\omega_n$ is the natural circular frequency of the n-th mode of piston.

Modal force $\ddot{f}_{p,n}$ is written by

$$\ddot{f}_{p,n} = \sum_{i=1}^{N_F} F_{Ai} \cdot \phi_{i,Ai,n} - \sum_{i=1}^{N_F} F_{Ei} \cdot \phi_{i,Ei,n} \quad (6)$$

(2) Equation of motion of cylinder liner

Liner vibration is induced by the piston impact force $F_{Ai}, F_{Ei}$. Vibration amplitude of cylinder liner $x_{Li}$ is expressed by the sum of eigen mode $\psi_n$.

$$x_{Li} = \sum_{n=1}^{N_{Li}} \psi_{i,n} \cdot b_n(t) \quad (7)$$

The equation motion of the n-th mode of liner $b_n$ is

$$\ddot{M}_{l,n} \dot{b}_n + 2\zeta_n \omega_n \dot{M}_{l,n} \dot{b}_n + \ddot{M}_{l,n} \omega_n^2 b_n = \ddot{f}_{l,n} \quad (8)$$

where $\ddot{M}_{l,n}$ is the modal mass, $\zeta_n$ is the damping ratio and $\omega_n$ is the natural circular frequency of the n-th mode of liner.

Modal force $\ddot{f}_{l,n}$ is given by

$$\ddot{f}_{l,n} = -\sum_{i=1}^{N_F} F_{Ai} \cdot \psi_{i,Ai,n} + \sum_{i=1}^{N_F} F_{Ei} \cdot \psi_{i,Ei,n} \quad (9)$$

Dynamic responses of piston and liner are obtained by numerical calculation of Eq. (1), (5), and (8). Impact force of Eq. (3) is also determined in the same procedure.

2.2 Evaluation of engine noise radiated from the engine block

Engine noise prediction requires the evaluation of vibratory response of engine block and its related acoustic radiation power. Authors \(^{(5),(6),(8)}\) have developed the numerical procedure to predict the vibratory response of engine block coupled with the rotating crankshaft and gear trains taking into account the gas combustion force, piston slap force, inertia force, fuel injection force, valve train’s opening and closing forces. Using this method, effect of changing of piston slap force on the engine noise is discussed in this paper.

Acoustic power $W(\omega)$ radiated from the engine block surface at the circular frequency $\omega$ is given by

$$W(\omega) = \rho c \sigma(\omega) <V^2(\omega)> S \quad (10)$$

where $<V^2(\omega)>$ is spatially averaged mean square velocity of engine block surface, $\rho c$ is the specific acoustic impedance of air, $\sigma(\omega)$ is the sound radiation efficiency determined by BEM (Boundary Element method), $S$ is the areas of the engine block surface.
3. Effect of piston profile

Effect of the piston skirt profile on the piston impact forces and engine noise is examined. Fig.2 shows the two types of piston having different profile of skirt that collides with the cylinder liner. Minimum clearance between piston and liner is assumed to be 50µm. Clearance of piston① is almost constant and clearance of piston② increases gradually from bottom to top of piston skirt. In this paper, piston profile is approximated by the sequence of 50 points at thrust and anti-thrust side and impact spring and dashpot are located at each point. Stiffness of impact spring is derived based on the Hertz contact theory and damping coefficient of dashpot is determined by the lubrication analysis of oil film.

Present method is applied to 4cycle and 4cylinders high speed diesel engine with displacement of 3.5 liter. Engine revolution speed is NE=2200rpm with no load condition and piston pin offset is assumed to be 0µm. Fig.3 shows the analytical results of wave form of piston slap force. As the number of impact points between liner and piston is 100 points and this number is too many to show the impact force of each point, sum of the 25 impact forces of upper and lower parts of piston are defined as F_AB_Upper, F_AB_Lower at the thrust side and F_CD_Upper, F_CD_Lower at anti-thrust side.

As shown in Fig.3, one may see large impact force in F_AB_Upper at the vicinity of combustion top dead center (crank angle \(\alpha=0^\circ\)) and there is no distinct difference among piston① and piston②. On the other hand, sharp impulsive force is seen in F_CD_Lower at the same timing and this impact force of piston② is larger than that of piston①. There is a sharp impulsive wave form in F_CD_Upper at the crank angle \(\alpha=380^\circ\) and this impact force of piston② is also larger than that of piston①. It is considered that change of piston impact location causes the difference of waveforms of upper and lower impact force induced by piston side forces and moment around piston pin.
Fig. 3 Piston impact force
Fig. 4 shows the narrow band frequency spectrum of upper impact force $F_{\text{AB,Upper}}$ and Fig. 5 also shows 1/3 octave band frequency spectrum of total impact forces. As frequency spectrum of impact force of piston① is smaller than that of piston② above the frequency of 1.25kHz, profile of piston① is supposed to be more effective to reduce the engine noise in the high frequency range. It is also reported in reference (7) that small clearance at the upper part of piston decreased the high frequency components of engine noise induced by piston slap.

![Fig.4 Frequency spectrum of upper impact force $F_{\text{AB,Upper}}$](image)

![Fig.5 Frequency spectrum of total impact force](image)

4. Abating the piston impact force by pin offset

Piston pin offset is common measure to reduce the engine noise induced by piston slap. As effect of piston pin offset on the piston slap and engine noise was discussed by use of the two points impact model on both side in the reference (2), new analytical method incorporating multipoint collision model is applied to evaluate the effect of piston pin offset in this paper. Change of impact force waveform and engine noise are numerically examined in the case that pin offset of piston① shown in Fig. 2 is changed from -1.0mm(thrust side) to 1.0mm(anti-thrust side). Fig. 6 shows the calculated results of piston impact force when the pin offset are -1.0mm and 0.0mm. Just after the combustion TDC (Top Dead Center α...
=0°), large amplitude of impact force is generated.

Fig6. Piston impact force vs. piston pin offset

(a) Pin offset $x_{p0} = -1.0\,\text{mm}$

(b) Pin offset $x_{p0} = 0.0\,\text{mm}$

Fig6. Piston impact force vs. piston pin offset
Fig. 7 shows the relation between this large amplitude of impact force and piston pin offset. When the piston pin is shifted to the major thrust side (negative offset), amplitude of impulsive force just after combustion TDC decreases. But this peak value is almost same, when piston pin was shifted to anti-thrust side (positive offset). Combustion pressure induced moment $F_g \cdot x_{po}$ and friction moment around piston pin $T_p$ are shown in Fig. 8 in the case that the pin offset are -0.5mm and -1.0mm.

![Fig.7 Maximum impact force vs. pin offset](image1)

![Fig.8 Pin offset and friction moment](image2)
If the pin offset is taken to be -0.5 to -1.0mm, combustion pressure induced moment $F_G \cdot x_{po}$ around piston pin becomes almost equal to the friction moment $T_p$. Therefore, if piston pin is offset to the thrust side, combustion pressure induced moment $F_G \cdot x_{po}$ can cancel the friction moment $T_p$ and it leads to mitigate the piston rotation that is one of the main causes of piston impact at the vicinity of TDC. Fig.9 shows the measured and calculated engine noise level changing with the piston pin offset. In this calculation, calculated piston slap force is employed as one of the exciting forces acting on the engine block coupled with the rotating crankshaft and gear trains. Tendency of the calculated engine noise that changes with the piston pin offset almost agrees with the measured results.

5. Conclusion

Theoretical procedure to predict piston slap was developed considering the precise piston profile and it was applied to evaluate the effect of piston profile and piston pin offset on the engine noise. It was clarified that decreasing the piston clearance of upper part reduced the high frequency components of engine noise. Moreover piston pin offset to the thrust side by 0.5~1.0mm mitigated the piston slap induced noise. These calculated results agreed well with the measured ones and availability of the present method was confirmed.

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