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LOW NOISE DESIGN OF DIESEL ENGINE INDUCED BY PISTON SLAP

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Abstract

This paper presents the theoretical procedure to predict the piston slap and its related noise radiation from the engine block. Using this method, the optimum value of the piston pin offset to reduce the impact force and engine noise is determined and compared with the measured one to evaluate the availability of this theoretical method. And it is revealed that piston impact force in the vicinity of the combustion top dead center is controlled by the friction moment around the piston pin and the piston pin offset moment induced by combustion pressure.

INTRODUCTION

Piston slap in an internal combustion engine is a transient impact phenomenon generated in the clearance between the piston and cylinder liner when the side force acting on the reciprocating piston changes its direction. And it is the one of the major sources of engine noise and liner cavitation [1]-[7]. Although decreasing the clearance between liner and piston is most effective to reduce the piston impact forces and engine noise, the clearance decrement should be precisely conducted to avoid the outbreak of piston-scuffing. Another measure to reduce the piston slap induced noise is the offset of the piston pin to the major thrust side.

This paper summarizes the analytical method to evaluate the impact force and engine noise induced by piston slap [8]. This method is applied to determine the optimum value of the piston pin offset to abate the piston slap and the calculated optimum value is compared with the measured one to confirm the availability of this method. And it is pointed out that piston movement in the vicinity of the combustion top dead center is governed by the friction moment around the piston pin and the additional piston pin offset moment induced by combustion pressure.

ANALYSIS OF PISTON SLAP

Analytical Model of Piston Slap

Fig.1 shows the coordinate system and collision model. An assumption is set forth that the piston and the liner collide at the 4 points of A, B, C and D. The liner is treated as an elastic structure because there are several resonance modes in the frequency range below 3 kHz in which the spectrum of engine noise becomes dominant. As the piston's first resonance frequency is higher than 3 kHz in the case of the high speed diesel engine, the piston can be treated as a rigid body having resilient springs at collision points. In addition a dash pot, representing the damping effect of the oil film, is assumed at each collision point.

Forces and Moments Acting on Piston and Liner

In the present method, following forces and moments shown in Fig.2 are assumed to act between the piston and the liner.

- F_{IX} : Inertia force in transverse direction due to piston motion
- F_{IY} : Inertia force in vertical direction due to piston motion
- T_I : Rotational moment of inertia around the pin due to piston motion
- F_g : Gas force
- F_l : Reaction force of connecting rod
- T_P : Moment around the piston pin due to friction force between the piston and the piston pin
- F_{rj} : Friction force in transverse direction between the j-th piston ring and the ring groove
- F_{qj} : Friction force in vertical direction between the j-th piston ring and the liner
- F_{f1}, F_{f2} : Friction force in vertical direction between the piston and the liner
- F_A, F_B, F_C, F_D : Impact force between the piston and the liner

In Fig.3, the coordinates of the piston pin are represented by $P(x_P, y_P)$, its rotational angle by θ_P, coordinates of the center of gravity of the piston by $G(x_G, y_G)$, and its rotational angle by θ_G.

Assuming a constant rotating angular velocity ω for the crank shaft, vertical displacement y_P is uniquely defined by the rotational angle of crank $\alpha = \omega t$.

Referring to Fig.3, vertical acceleration of the piston pin $\ddot{y}_P$, when the piston pin offset x_{P0} is considered, is given approximately as follows:

$$\ddot{y}_P = -r\omega^2\left[\cos(\omega t-\gamma_0)+\frac{r\ell^2\cos^2(\omega t-\gamma_0)-A\left(\ell^2-A^2\right)\sin(\omega t-\gamma_0)}{\left(\ell^2-A^2\right)^{3/2}}\right] \qquad (1)$$

where l is the length of the connecting rod, r is the crank radius and

$A = r\sin(\omega t-\gamma_0)-(x_P+x_{P0})$

Reaction force from the connecting rod F_ℓ is determined by the equilibrium condition among vertical forces acting on the piston.

$$F_\ell = F_{\ell 1} - \frac{m_P L_X}{\cos\beta}\ddot{\theta}_P \quad \text{where } F_{\ell 1} = \frac{\left(m_P + m_{rg} + m_r\right)\ddot{y}_P - F_g - \sum_{i=1}^{Nr} F_{qi} - F_{f1} - F_{f2}}{\cos\beta} \tag{2}$$

The first term of Eq.(2) is by determined by the rotational angle of the crank while the second term being governed by the rotating motion of the piston. Reaction force of the connecting rod F_ℓ acts on the bearing of the piston pin. By representing the friction coefficient between the pin shaft and the pin bearing by μ_P, moment around the pin T_P is expressed by

$$T_P = \mathrm{sgn}(\dot{\varphi})\mu_P R_P F_\ell \tag{3}$$

where $\begin{cases} \dot{\varphi} = \dot{\beta} - \dot{\theta}_P \quad R_P\text{: Radius of piston pin} \\ \mathrm{sgn}(\dot{\varphi}) = \begin{cases} 1 & \cdots\cdots \quad \dot{\varphi} > 0 \\ -1 & \cdots\cdots \quad \dot{\varphi} < 0 \end{cases} \end{cases}$

Impact force F_S between the piston and the liner, at collision point S, is given by Eq.(4) using spring constant k_S and the damping coefficient c_S that simulate the dynamic stiffness of the piston and the damping effect of oil film.

$$F_S = -\delta_S\left\{k_S\left(x_{PS} - x_{LS}\right) + c_S\left(\dot{x}_{PS} - \dot{x}_{LS}\right)\right\} \tag{4}$$

where $\delta_S = \begin{cases} 1 - - - \text{colliding} \\ 0 - - - \text{not\_colliding} \end{cases}$

Coupled Vibration of Piston and Liner

Equation of Motion of Piston System

The piston is regarded as a rigid body having a resilient spring at the collision point with the liner. In addition, vertical motion of the piston pin is uniquely determined by the rotational angle of the crank. Therefore, transverse displacement x_P at piston pin position, and rotating angle θ_P around the piston pin, are considered.

The equation of motion for the piston is given by the following equation according to the equilibrium of forces acting on the piston as above mentioned.

$$\begin{bmatrix} m_p + m_r & m_P\left(L_Y - L_X \tan\beta\right) \\ m_P L_Y & I_P \end{bmatrix} \begin{bmatrix} \ddot{x}_P \\ \ddot{\theta}_P \end{bmatrix} = \begin{bmatrix} f_x \\ f_\theta \end{bmatrix} \tag{5}$$

where $L_X = x_G - x_P, L_y = y_G - y_P$

$$f_X = -F_{\ell 1}\sin\beta + \sum_{j=1}^{Nr} F_{rj} + \sum_{S}^{A\sim D} F_S \tag{6}$$

$$f_\theta = m_P L_X \ddot{y}_P - F_g\left(x_t - x_P\right) - \sum_{j=1}^{2} F_{fj}\left(x_{fj} - x_P\right) - T_P + \sum_{j=1}^{Nr} F_{rj}\left(y_{rj} - y_P\right) + \sum_{S}^{A\sim D} F_S\left(y_S - y_P\right) \tag{7}$$

N_r :Number of rings
x_t : x coordinate of the upper center of piston
x_{fj} : x coordinate at position where friction force between the piston and the liner is acting
y_{rj} : y coordinate of the j-th ring

Equation of Motion for Liner System

As the liner and engine block are considered as an assembly, the effect of the engine block must be taken into account when evaluating the vibratory response of the liner. The equation of motion for the liner system, where impact force $\mathbf{f}$ with piston is acting, is expressed as follows by using mass matrix $\mathbf{M}$, damping matrix $\mathbf{C}$ and stiffness matrix $\mathbf{K}$, for the liner system.

$$\mathbf{M\ddot{u}} + \mathbf{C\dot{u}} + \mathbf{Ku} = \mathbf{f} \tag{8}$$

where $\mathbf{u}$ represents the displacement vector of liner system.

Dynamic characteristics of the liner system, including engine block, can be obtained by the FEM calculation or vibration test. Fig.4 shows the measured result of the driving point mobility for the liner installed in the engine block. From the result of such a vibration test, the modal parameters of the liner system (natural circular frequency ω_n, normal mode $\boldsymbol{\varphi}_n$, effective mass $\tilde{m}_n$, effective damping ratio ς_n, suffixes n represents the relevant quantity relates to the n-th mode) can be identified by using a curve fit technique shown in Reference[9].

Vibratory displacement $\mathbf{u}$ of the liner system, impacted by the piston, is represented by a linear combination of the normal modes $\boldsymbol{\varphi}_n$ from the first to N-th mode, as follows.

$$\mathbf{u} = \boldsymbol{\varphi}_1 a_1 + \cdots\cdots + \boldsymbol{\varphi}_N a_N = \sum_{n=1}^{N} \boldsymbol{\varphi}_n a_n \tag{9}$$

where a_n is the modal response of the n-th mode.
The equation of motion for the n-th mode is represented as follows:

$$\ddot{a}_n + 2\zeta_n \omega_n \dot{a}_n + \omega_n{}^2 a_n = \sum_{S}^{A\sim D} \phi_n(y_S) F_S / \tilde{m}_n \tag{10}$$

In the above, the n-th normal mode $\phi_n(y_s)$ at impact point S changes with the crank angle.
The piston and the liner are subject to coupled vibration via impact force F_s. The equations of motion for the piston and the liner, namely Eqs.(5) and (10), are reduced to simultaneous ordinary differential equations. At each of the impact points considered, numerical integration is carried out taking into account the impact conditions. Thus, the vibratory responses of the liner and the piston can be obtained at each crank angle, together with the time histories of impact force.

PREDICTION OF ENGINE NOISE

Vibration Response of Engine Block Coupled with Rotating Crankshaft

Reference [8], describes theoretical procedure to calculate the vibratory response of engine block coupled with the rotating crankshaft considering the dynamic characteristics of each structure, stiffness of the oil film and local structure at main bearings and exciting forces such as combustion pressure, piston slap and inertia forces. Spatially averaged mean square velocity $\langle V^2(\omega)\rangle$ is determined by use of the calculated vibration response of the crankcase surface.

Noise Radiation from Engine Block

Acoustic power W radiated from the engine block surface at the circular frequency ω is given by

$$W(\omega) = \rho c\, \sigma \langle V^2(\omega)\rangle S \tag{11}$$

where ρc is the acoustic characteristic impedance of air, σ is the sound radiation efficiency, S is the areas of the engine block surface. In addition, the sound radiation efficiencies are determined on the each eigen mode of engine block [10]

EFFECT OF PISTON PIN OFFSET ON IMPACT FORCE AND NOISE

Calculation of Piston Impact Forces

The clearance between the piston and liner was estimated based on the measured temperature of the piston. Fig.5～Fig.7 show the time history of impact forces in the case of pin offset x_{p0}=0mm , x_{po}= -0.5mm (thrust side), and x_{po} =+0.5mm (minor thrust side). In Fig.5 (x_{po} =0mm), the upper impact point A where the contact stiffness is harder than that of lower point B collides first and then lower point B collides, because the friction moment T_p acts around piston pin counterclockwise in the vicinity of the combustion TDC(top dead center). In Fig.6 (x_{po} = -0.5mm; thrust side), the lower point B collides first and the upper point A collides next due to the additional moment $F_g* x_{po}$ that acts around piston pin clockwise and cancels the friction moment. On the other hand in Fig.7 (x_{po} =+0.5mm ; minor thrust side), severe impact occurs first at the upper point A and then lower point B collides due to the additional moment $F_g* x_{po}$ that acts around piston pin counterclockwise.

Peak Value of Impact Force and Engine Noise

The frequency spectra of impact forces F_A, F_B, F_C, F_D are shown in Fig. 8. Impact force F_A at the upper impact point of major thrust side is much larger than those of other forces and the magnitude of its frequency components depends on the peak value of the wave form in the vicinity of the combustion TDC.

Fig.9 shows the relations between the piston pin offset x_{p0} and the peak value of the impact force F_A, engine noise level L_p. The peak value of the impact force F_A has a fair correlation with the engine noise level and the piston pin offset to the major thrust side seems to reduce the engine noise level. Calculated result implies that the peak value of the impact force F_A and the engine noise level L_p reach the minimum value at the piston offset x_{po}= -0.5mm and noise reduction of 1～2dB is obtained compared with the case of pin offset x_{po}=0mm. This is due to the cancellation of the friction moment around piston pin by the additional pin offset moment $F_g* x_{po}$. In such condition, the rotational movement of the piston becomes small and therefore its related impact forces decrease.

Fig.9 also shows the measured engine noise level that changes with the piston pin offset. Tendency of the change of the engine noise level with the piston pin offset is almost same between the calculated and measured results. Although the engine noise is generated by the many exciting forces other than piston slap, piston slap induced noise has the dominant role in the comparative higher frequency range and therefore piston pin offset leads to the engine noise reduction.

Discussion of the Optimum Pin Offset

The translational motion of the piston is governed by the side force $F_s=F_l\sin\beta$ and the rotational motion of the piston is controlled by the friction moment T_p and additional pin offset moment $F_g * x_{po}$. Fig.10 shows the friction moment T_p and additional pin offset moment $F_g * x_{po}$ changing with the crank angle in the vicinity of the combustion TDC. In Fig.10, additional pin offset moments $F_g * x_{po}$ are shown in the case of x_{po}=0.5 and 1.0mm and these values are almost the same level as the friction moment T_P. This implies that impact force induced by the rotational moment can be decreased by adjusting the value of the piston pin offset.

Fig.11 describes the relation of the summation of the rotational moment around piston pin $T_P+ F_g * x_{po}$ and pin offset value x_{po}. If one offsets the piston pin to the thrust side by the value of 0.6mm, rotational moment around the piston pin is almost zero and one can reduce the piston impact force. This optimum value of x_{po}=0.6mm fairly agrees with the measured optimum value x_{po}=0.5mm in Fig.9.

CONCLUSION

This paper presented the theoretical procedure to predict the piston slap impact force considering the dynamic properties of the piston and the liner. And using this method, optimum value of the piston pin offset to reduce the engine noise was examined and compared with the measured data. As the piston slap impact force in the vicinity of the combustion TDC is controlled by the rotational moment around the piston pin, abating the rotational moment is essential to reduce the engine noise. This paper offered the simple way to determine the optimum value of piston pin offset to cancel the friction moment T_p around the piston pin by the additional pin offset moment $F_g * x_{po}$. And availability of this method was confirmed by the analysis and measurement.

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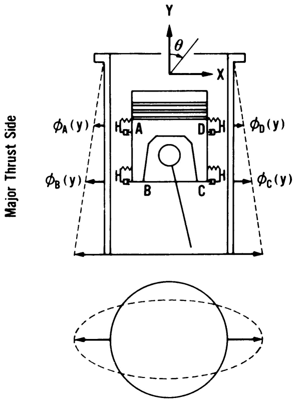

Figure-1 Analytical Model

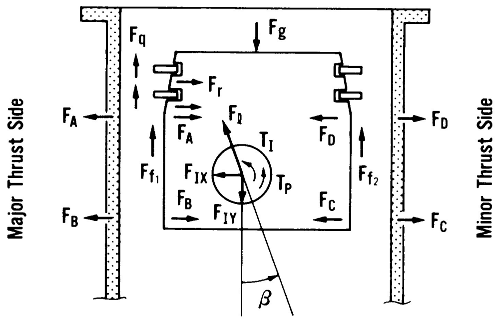

Figure-2 Forces and Moments acting on the piston

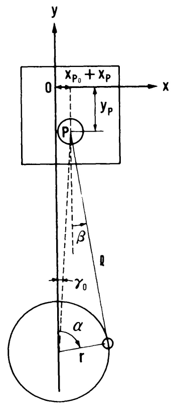

Figure-3 Piston pin offset

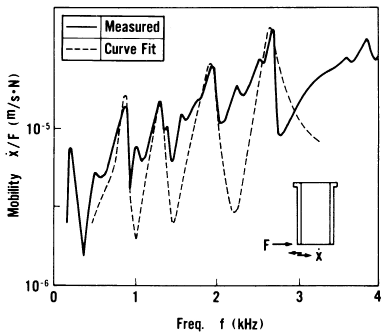

Figure-4 Mobility of cylinder liner

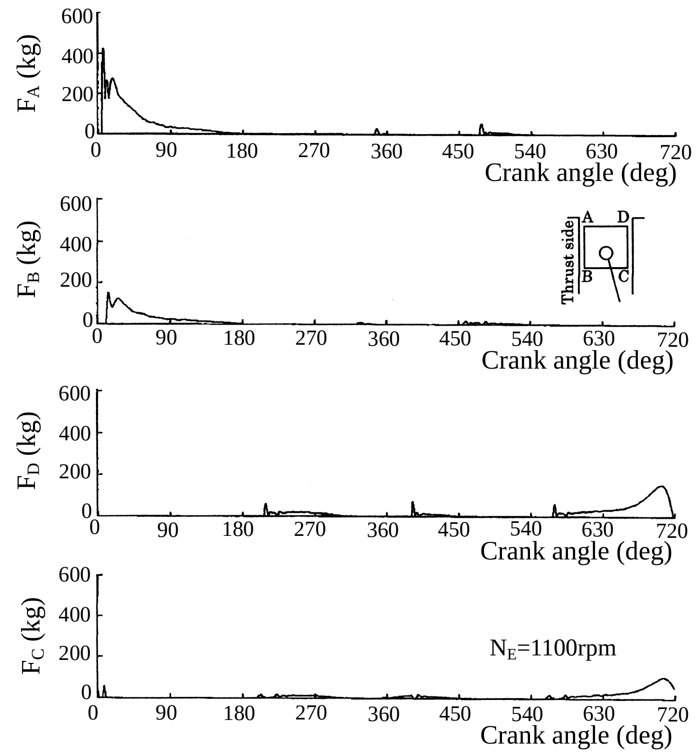

Figure-5 Piston impact force (x_{P0}=0mm)

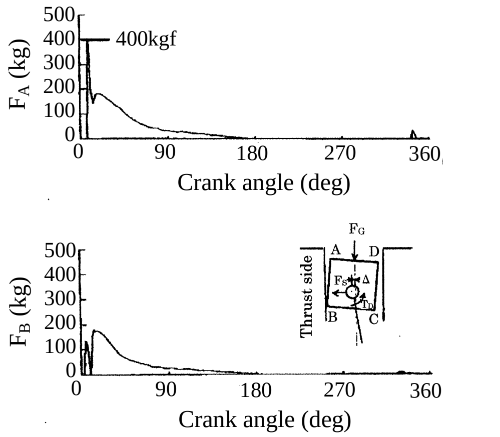

Figure-6 Piston impact force (x_{P0}= -0.5mm)

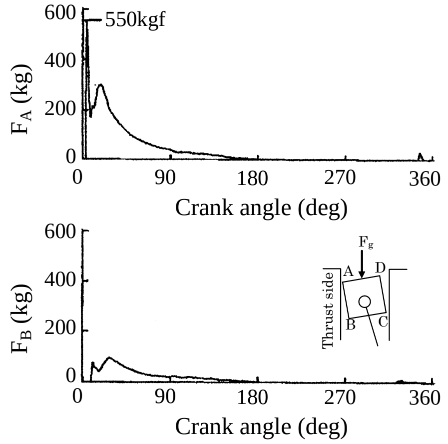

Figure-7 Piston impact force (x_{P0}=+0.5mm)

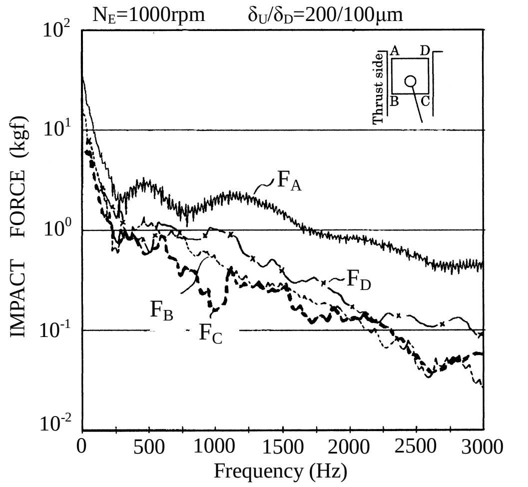

Figure-8 Frequency spectra of impact forces

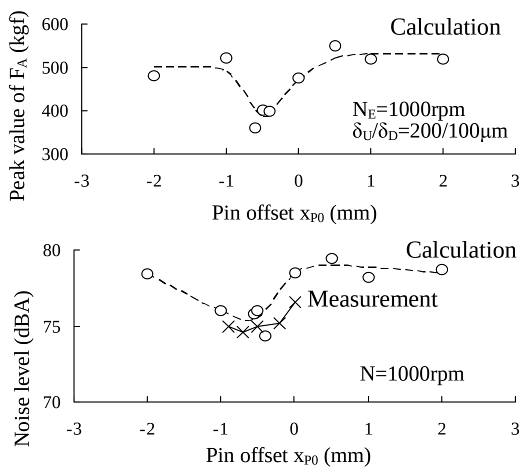

Figure-9 Effect of piston pin offset

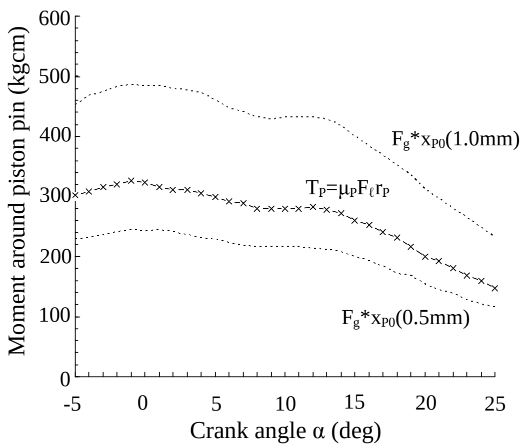

Figure-10 Moments around piston pin

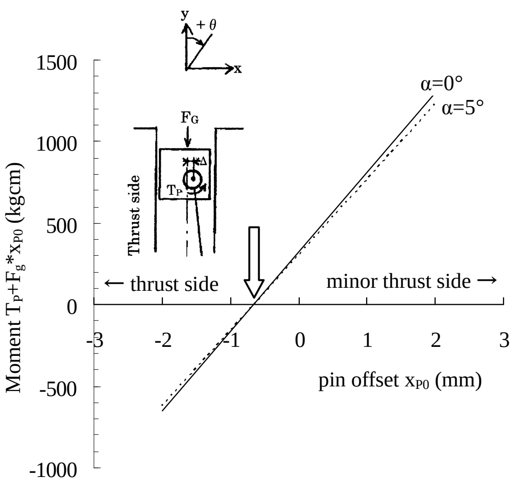

Figure-11 Total moment around piston pin