

# ANALYSIS OF VIBRATORY RESPONSE AND NOISE RADIATION OF ENGINE BLOCK COUPLED WITH THE ROTATING CRANKSHAFT AND GEAR TRAIN SYSTEM

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### Abstract

This paper presents the theoretical procedure to predict the vibratory response and radiated noise of the engine block coupled with the rotating crankshaft and gear train shafts which drives the fuel injection pump and valve system. The exciting forces acting on the engine block and shaft system are combustion pressure, inertia forces of the moving parts, piston slap forces, fuel injection pressure and valve driving force and torque. Theoretical procedures consist of the following four steps; (1) Dynamic characteristics of the engine block and shafts are determined separately by FEM or experimental modal analysis. (2) Normal mode expansion technique is employed to derive the equation of motion of the total system in which rotating shafts with gear train are combined to the engine block by the oil film and contact stiffness. (3) The time histories of the vibratory response of the engine block and rotating shafts are calculated by the numerical integration technique. (4) Engine noise radiated from the engine block surface is evaluated using the spatially averaged mean squared velocity and acoustic radiation efficiencies of the engine block. This method is applied to estimate the effect of the backlash of the gear train on the engine block vibration and radiated noise.

# **1. INTRODUCTION**

Engine noise has become one of the major factors of the marketability of the medium-to-high speed diesel engine along with the low emission control and low fuel consumption. Figure 1 shows the generation mechanism of the engine noise. For abating engine noise, there are many subjects to be studied such as the low noise structural design of the crankcase [1], control of the mechanical exciting forces like piston slap [2],[3] and main bearing impacts[4]. Noise control of the internal combustion engine in the early design stage enjoins the numerical procedure to estimate exciting forces, vibratory behaviour of the engine block and noise radiation properties. And recently the fuel injection pump driven by the crankshaft via gear trains is considered one of the major noise sources of the internal combustion engine because it injects instantly the fuel fluid at high pressure and induces the impulsive vibration of the gear trains and pump structure. This paper presents a numerical approach to predict the vibratory response of the engine block

coupled with the rotating crankshaft and gear train system considering the dynamic characteristics of each structure, stiffness of the oil film and structure at bearings and exciting forces such as combustion pressure, inertia force, piston slap and fuel injection pressure and so on. Numerical simulation offers the wave form of the acceleration of the engine block and rotating shaft system. Engine noise radiated from the engine block is estimated by use of the spatially averaged mean squared velocity and acoustic radiation efficiencies determined by the

Boundary Element Method . In this paper, the effect of the gear backlash of the timing gear trains on the engine noise is discussed.



Figure 1. Generation mechanism of the engine noise.

### **2. THEORY**

### **2.1 Analytical Model**

Figure 2 shows the analytical model in which the rotating crankshaft drives the fuel injection pump and valve system via the timing gear trains. These rotating shafts are installed into the engine blocks which are supported by the resilient mounts and connecting points between the engine block and shaft system are idealized by the spring and dashpot, the values of which are derived from the dynamic stiffness of the oil film and structural stiffness at the bearings. The exciting forces such as the combustion pressure, inertia force, piston slap impact force, fuel injection pressure and opening and closing force of the valve train act on the engine block and

the rotating shafts simultaneously. In Fig. 2, suffix 0, 1,2,3,4 denote the engine block, crankshaft, intermediate gear, fuel injection pump shaft, valve train shaft respectively. In the gear train system shown in Fig. 3, gears are connected by the spring and dashpot that corresponds to the dynamic stiffness of the gear tooth and each mating point has a clearance so called backlash. In the case of the helical gear train system, axial forces are considered together with the radial one.



Figure 2. Analytical model of the engine block and rotating shaft system.



Figure 3. Generation mechanism of the engine noise.

# 2.2 Equations of motion of the coupled system

Equations of motion of the engine block and gear train shafts are described in the stationary coordinate because the dynamic characteristics of the gear train shafts are considered to be approximately isotropic. As the dynamic property of the crankshaft is anisotropic, eigen mode

shape of the crankshaft observed in the stationary coordinate changes with the crank angle when the crankshaft rotates at the constant angular velocity  $\omega_0$ . Therefore, the equilibrium equation of the crankshaft is described in the rotating coordinate UVW which is fixed to the rotating crankshaft.

$$\mathbf{M}_{1}\ddot{\mathbf{u}} + (\mathbf{C}_{1} + \Delta \mathbf{C}_{1})\dot{\mathbf{u}} + (\mathbf{K}_{1} + \Delta \mathbf{K}_{1})\mathbf{u} = \mathbf{T}^{t}\mathbf{F}_{B1} + \mathbf{T}^{t}\mathbf{F}_{G1} + \mathbf{F}_{C} + \mathbf{F}_{M}$$
(1)

where  $M_1$ ,  $C_1$ ,  $K_1$  are mass, damping, and stiffness matrices of the crankshaft and  $\Delta C_1$ ,  $\Delta K_1$  are related to the Coriolis' force and centrifugal effect. Displacement of the crankshaft u is determined in the rotational coordinate. External forces to be considered are the reaction force of the main bearing impact  $F_{B1}$  and mating force with intermediate gear  $F_{G1}$ , gas and inertia force  $F_C$  acting on the crankpin and the centrifugal force  $F_M$ . T is the transform matrix from the stationary coordinate to the rotating coordinate[4],[5]. Main bearing impact force  $F_{B1}$  and gear mating force  $F_{G1}$  are expressed by the following equations

$$T^{t}F_{B1} = -K_{11}^{B}u - C_{11}^{B}u + K_{10}^{B}x_{0} + C_{10}^{B}x_{0}$$
$$T^{t}F_{G1} = -K_{11}^{G}u - C_{11}^{G}u + K_{12}^{G}x_{0} + C_{12}^{G}x_{0}$$
(2)

Suffix B and G represent the values related to the bearing and gear. Finally equation of motion of the coupled system is given by

$$\mathbf{MX} + \mathbf{CX} + \mathbf{KX} = \mathbf{F}$$
(3)  
$$\mathbf{X} = \begin{bmatrix} \ddot{\mathbf{x}}_{0} \\ \ddot{\mathbf{u}}_{1} \\ \ddot{\mathbf{x}}_{2} \\ \ddot{\mathbf{x}}_{3} \\ \ddot{\mathbf{x}}_{4} \end{bmatrix}, \quad \mathbf{M} = \begin{bmatrix} \mathbf{M}_{0} \\ \mathbf{M}_{1} \\ \mathbf{M}_{2} \\ \mathbf{M}_{3} \\ \mathbf{M}_{4} \end{bmatrix}, \quad \mathbf{F} = \begin{bmatrix} \mathbf{F}_{s0} + \mathbf{F}_{c0} + \mathbf{F}_{s3} + \mathbf{F}_{c3} + \mathbf{F}_{s4} + \mathbf{F}_{c4} \\ \mathbf{F}_{c} + \mathbf{F}_{M} \\ \mathbf{0} \\ \mathbf{F}_{c3} \\ \mathbf{F}_{c4} \end{bmatrix}$$
$$\begin{bmatrix} \mathbf{C}_{0} + \mathbf{C}_{00}^{\mathsf{B}} + \mathbf{C}_{\mathsf{E}} & -\mathbf{C}_{01}^{\mathsf{B}} & -\mathbf{C}_{02}^{\mathsf{B}} & -\mathbf{C}_{03}^{\mathsf{B}} & -\mathbf{C}_{04}^{\mathsf{B}} \end{bmatrix}$$

$$\mathbf{C} = \begin{bmatrix} -\mathbf{C}_{10}^{B} & \mathbf{C}_{1} + \Delta\mathbf{C}_{1} + \mathbf{C}_{11}^{B} + \mathbf{C}_{11}^{G} & -\mathbf{C}_{12}^{G} & \mathbf{0} & \mathbf{0} \\ -\mathbf{C}_{20}^{B} & -\mathbf{C}_{21}^{G} & \mathbf{C}_{2} + \mathbf{C}_{22}^{B} + \mathbf{C}_{22}^{G} & -\mathbf{C}_{23}^{G} & -\mathbf{C}_{24}^{G} \\ -\mathbf{C}_{30}^{B} & \mathbf{0} & -\mathbf{C}_{32}^{G} & \mathbf{C}_{3} + \mathbf{C}_{33}^{B} + \mathbf{C}_{33}^{G} & \mathbf{0} \\ -\mathbf{C}_{40}^{B} & \mathbf{0} & -\mathbf{C}_{42}^{G} & \mathbf{0} & \mathbf{C}_{4} + \mathbf{C}_{44}^{B} + \mathbf{C}_{44}^{G} \end{bmatrix}$$

$$\mathbf{K} = \begin{bmatrix} \mathbf{K}_{0} + \mathbf{K}_{00}^{B} + \mathbf{K}_{E} & -\mathbf{K}_{01}^{B} & -\mathbf{K}_{02}^{B} & -\mathbf{K}_{03}^{B} & -\mathbf{K}_{04}^{B} \\ -\mathbf{K}_{10}^{B} & \mathbf{K}_{1} + \Delta \mathbf{K}_{1} + \mathbf{K}_{11}^{B} + \mathbf{K}_{11}^{G} & -\mathbf{K}_{12}^{G} & \mathbf{0} & \mathbf{0} \\ -\mathbf{K}_{20}^{B} & -\mathbf{K}_{21}^{G} & \mathbf{K}_{2} + \mathbf{K}_{22}^{B} + \mathbf{K}_{22}^{G} & -\mathbf{K}_{23}^{G} & -\mathbf{K}_{24}^{G} \\ -\mathbf{K}_{30}^{B} & \mathbf{0} & -\mathbf{K}_{32}^{G} & \mathbf{K}_{3} + \mathbf{K}_{33}^{B} + \mathbf{K}_{33}^{G} & \mathbf{0} \\ -\mathbf{K}_{40}^{B} & \mathbf{0} & -\mathbf{K}_{42}^{G} & \mathbf{0} & \mathbf{K}_{4} + \mathbf{K}_{44}^{B} + \mathbf{K}_{44}^{G} \end{bmatrix}$$

$$(4)$$

#### 2.3 Modal analysis

Vibration displacement **u** of the rotating crankshaft observed in the rotating coordinate is expressed by the linear combination of the normal mode  $\boldsymbol{\phi}_{1n}$  which are determined in the stationary coordinate. Vibration displacement of the engine block and gear train system  $\mathbf{x}_q$ (q=0,2,3,4) are also expressed by the normal mode  $\boldsymbol{\phi}_{qn}$  determined in the stationary coordinate.

$$\mathbf{u}_{1} = \sum_{n=1}^{N} \boldsymbol{\varphi}_{1n} \mathbf{a}_{n} = \boldsymbol{\Phi}_{1} \mathbf{a}_{1} ,$$

$$\mathbf{x}_{q} = \sum_{n=1}^{N} \boldsymbol{\varphi}_{qn} \mathbf{a}_{n} = \boldsymbol{\Phi}_{q} \mathbf{a}_{q} \qquad q=0, 2, 3, 4 \qquad (5)$$

where  $\mathbf{a}_q = (a_{q1}, a_{q2}, \cdots, a_{qn})^t$  is the modal response vector and  $\mathbf{\Phi}_q = [\mathbf{\phi}_{q1}, \mathbf{\phi}_{q2}, \cdots, \mathbf{\phi}_{qn}]$  is the eigen mode shape matrix of the  $q_{th}$  structure. Substituting Eq.(4) into Eq.(3) and multiplying  $\mathbf{\Phi}_q t$  from the left side, one can rewrite the equation of motion in the modal coordinate as follows

$$\tilde{\mathbf{M}}\ddot{\mathbf{a}}+\tilde{\mathbf{C}}\dot{\mathbf{a}}+\tilde{\mathbf{K}}\mathbf{a}=\tilde{\mathbf{F}}$$
,  $\mathbf{a}=(\mathbf{a}_0,\mathbf{a}_1,\mathbf{a}_2,\mathbf{a}_3,\mathbf{a}_4)^{\mathrm{t}}$  (6)

As the number of the degrees of the freedom in Eq. (5) becomes the total number of the modes of the engine block and rotating shafts, numerical calculation in time domain is easy to carry out.

#### 2.4 Estimation of the engine noise spectrum

This analytical method yields detailed vibration velocity of the engine block surface. Spatially averaged mean square velocity of the engine block surface  $\langle V^2(\omega) \rangle$  is

$$\langle \mathbf{V}^{2}(\omega) \rangle = \frac{1}{S} \iint_{S} |\mathbf{V}_{e}(\omega)|^{2} dS, \qquad \mathbf{V}_{e}(\omega) = \sum_{n=1}^{N} \boldsymbol{\varphi}_{en} a_{n}(\omega)$$
(7)

Acoustic radiation power  $W(\omega)$  is determined by [6]

$$W(\omega) = \rho c < V^{2}(\omega) > \sigma S$$
 (8)

where  $\rho$  c is the specific acoustic impedance of air,  $\sigma$  is the acoustic radiation efficiency of the engine block evaluated by the Boundary Element Method, S is the surface area of the engine block.

# **3. CALCULATED RESULTS**

### 3.1 Impulsive vibration of the timing gear trains

Figure 4 shows calculated results of the time histories of the fluctuating mating force between the intermediate gear and driving gear of fuel injection pump at the engine revolution speed  $N_E$ =1000rpm. When the gear backlash is zero, continuous vibratory mating force, amplitude of which is about 200N, is induced by the fluctuating angular displacement of the crankshaft and impulsive torque of the fuel injection pump at the vicinity of the combustion top dead centre (TDC). On the other hand, a series of impulsive mating force, which amplitude is roughly 800N,

originates at the vicinity of the combustion TDC when the gear backlash is  $\delta = \pm 40 \mu m$ ,





# 3.2 Change of the gear backlash and its effect on the engine noise

Decreasing the gear backlash is often employed to reduce the engine noise. Figure 5 shows calculated results of the variation of the engine noise level when the gear backlash changes. Decreasing the gear backlash is effective way to reduce the engine noise at the low engine

revolution speed  $N_E$ =1000rpm. But in the case of the high engine revolution speed  $N_E$ =2200rpm, decreasing of the gear backlash is not valid to abate the engine noise level. This tendency is confirmed by the measured results of the actual engine with similar specification as shown in Fig. 6. This is because the angular displacement of the gear train driven by the crankshaft is large when the engine revolution speed is low and the angular displacement becomes small at the high engine revolution speed as shown in Fig. 7.



Figure 5. Gear backlash and engine noise (Calculation).

#### **4. CONCLUSION**

Analytical procedure to evaluate the vibration response and radiated noise of the engine block coupled with the rotating crankshaft and gear train system was developed. This method was applied to evaluate the effect of the gear backlash onto the engine noise. The calculated result shows that the decreasing of the gear backlash can reduce the engine noise level at the low engine rotational speed and attain little noise reduction at the high revolution speed. This calculated result almost agrees with the measured one and the availability of this theoretical procedure has been confirmed.

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Figure 6. Gear backlash and engine noise (Measured)



Figure 7. Fluctuation of the angular displacement of the rotating crankshaft.