Study on the noise and vibration of engine block coupled with the rotating crankshaft and gear train
(Effect of the torsional vibration of crankshaft)

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Abstract
As the small-sized high speed diesel engine used for construction machinery and generator is operated under continuous heavy load condition, a gear train is employed to drive the fuel injection pump and valve train. When a loading torque is small at the idling condition, the gear tooth separation and impact are caused by the fluctuating torque and they increase the engine noise level. Particularly when the torsional vibration of the crankshaft occurs and its rotational amplitude is magnified at a specified rotational speed, the gear impact is intensified and it leads to increase of engine noise. Author has developed the theoretical procedure using FEM, modal analysis technique and BEM 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. This method is applied to reveal the effect of torsional vibration of crankshaft on the gear impact force and engine noise of the four-stroke four-cylinder engine. Countermeasure to reduce vibration and noise by changing the location of the gear train from pulley side to flywheel side of the crankshaft is evaluated and its result indicates the prospects which can reduce the radiated noise from the engine block.

Key words : Internal combustion engine, Vibration, Noise, Modal analysis, FEM, BEM

1. Introduction

Diesel engine used for the power generation, forklift and construction machinery is required to perform the continuous and heavy load operation, therefore the gear train driven by crankshaft is employed to run the fuel injection pump and valve system. As the driving torque of rotating components such as injection pump is small at the idling condition, separation of gear tooth occurs and its impact force increases the engine noise level. When the torsional vibration of the crankshaft is generated, the rotational displacement of crank gear is magnified and gear impact between the crank gear and intermediated gear is intensified. For abating engine noise, there have been many researches to control the mechanical exciting force like piston slap (Ohta, et al., 1987, Koizumi, et al., 2001, 2002, Suganuma, et al., 2006) and main bearing impact of crankshaft (Ohta, et al., 1993) and to attain the low noise structural design of the crankcase (Nakamura, et al., 1990, Sohn, et al., 1991). But there have been only a few researches on the coupled vibration of engine block and rotating crankshaft because the vibratory characteristics of the crankshaft is complicated and its eigen mode shape observed in the stationary coordinate changes with the crank angle. Although recently commercial software based on FEM to calculate the dynamic behaviour of engine structure has become available, it seems not to be easy task to carry out the detailed dynamic analysis of entire engine structure taking account the effect of the moving, rotating and rattle of each engine components. Author has developed a theoretical 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 local structure at bearings and exciting forces.
such as combustion pressure, inertia force, piston slap and fuel injection pressure and so on (Ohta, et al., 1994, 2002, 2011). Numerical simulation offers the time histories of the acceleration of the engine block and rotating shaft system. Engine noise radiated from the engine block is estimated by use of the Boundary Element Method (BEM). These calculated results were compared with the measured data and the availability of these numerical procedure was confirmed. This paper examines the effect of the torsional vibration of crankshaft on the gear impact force and engine noise of the four-stroke four-cylinder engine. As the gear train is equipped at the pulley side of the crankshaft and the torsional vibration mode of the crankshaft is enlarged at the pulley side than flywheel side, a countermeasure to reduce vibration and noise by changing the location of the gear train from the pulley side to the flywheel side is discussed.

Nomenclature

In this paper, suffix 0, 1, 2, 3, 4 represent the engine block, the crankshaft, the intermediate gear shaft, the fuel injection pump shaft and the valve train shaft. Suffix $B$ shows the bearing, $G$ shows the gear and $E$ is the engine mount. Forces of $F_M$ and $F_C$ are defined in the rotating coordinate and other forces are described in the stationary coordinate.

\[ a = (a_0, a_1, a_2, a_3, a_4)^T : \text{Modal response vector} \]
\[ C : \text{Damping matrix} \]
\[ F_{B1} \sim F_{B4} : \text{Bearing forces acting on crankshaft, intermediate gear shaft, injection pump shaft and valve train shaft} \]
\[ F_C : \text{Force acting on crankpin caused by combustion pressure and inertia force of piston and connecting rod} \]
\[ F_{C3} : \text{Force acting on injection pump caused by injection pressure including side force} \]
\[ F'_{C3} : \text{Force and torque acting on the shaft of injection pump} \]
\[ F_{C4} : \text{Force acting on engine block caused by valve opening and closing force} \]
\[ F'_{C4} : \text{Force and torque acting on valve train shaft caused by valve opening and closing force} \]
\[ F_{EB} : \text{Reaction force from engine mount} \]
\[ F_{GB} : \text{Gas combustion force} \]
\[ F_{GI} : \text{Meshing force between crankshaft gear and the intermediate gear} \]
\[ F_{G2} : \text{Meshing forces acting on the intermediate gear} \]
\[ F_{G3} : \text{Meshing force between injection pump gear and intermediate gear} \]
\[ F_{G4} : \text{Meshing force between valve train gear and intermediate gear} \]
\[ T : \text{Transformation matrix from the rotating coordinate to the stationary coordinate} \]
\[ F_M : \text{Centrifugal force acting on the crankshaft} \]
\[ F_{SB} : \text{Piston slap force} \]
\[ h_{lm}, g_{lm} : \text{Influence coefficients in BEM} \]
\[ K : \text{Stiffness matrix} \]
\[ k=\omega/c : \text{Wave number, } c \text{ is speed of sound.} \]
\[ M : \text{Mass matrix} \]
\[ p : \text{Sound pressure, } p^* : \text{Complex conjugate of sound pressure} \]
\[ x : \text{Vibration displacement defined in the stationary coordinate (Translation + Rotation)} \]
\[ u : \text{Vibration displacement defined in the rotating coordinate} \]
\[ \zeta_{qn} : \text{Damping ratio of the n-th mode of the structure } q \ (q=0,1,2,3,4) \]
\[ \omega : \text{Angular frequency} \]
\[ \Phi_{gn} : \text{Mode shape of the n-th mode of the structure } q \]
\[ \Phi_q = [\Phi_{g0}, \Phi_{g1}, \Phi_{g2}, \Phi_{g3}^T] : \text{Mode shape matrix of the structure } q \]
\[ \mathbf{M}, \mathbf{K}, \mathbf{C} : \text{Modal mass, damping and stiffness matrix} \]
\[ \mathbf{F} : \text{Modal force vector} \]

2. Analytical model

2.1 Analytical model of engine block and rotating shafts

Figure 1 shows the analytical model of engine system in which the rotating crankshaft drives the fuel injection pump and valve system through the timing gear train. These rotating shafts are installed into the engine blocks which are supported by the resilient mounts. Connecting points between the engine block and shaft system are idealized by the spring and viscous damping, the values of which are derived from the dynamic stiffness of the oil film and local
structural stiffness at the bearings. Finite Element Model of engine block shown in Fig.1 consists of crankcase, cylinder head, front plate, gear casing and inline fuel injection pump. As major frequency components of engine noise of the high speed diesel engine which is investigated in this study are below 3kHz (Ohta, et al., 1994), vibratory characteristics of each engine components are taken into account below 4kHz.

2.2 Exciting forces

As shown in Fig.2, the exciting forces such as the combustion pressure $F_{\text{Gb}}$, piston slap impact force $F_{\text{sb}}$, fuel injection pressure induced force and moment $F_{\text{C3}}$, opening and closing force and moment of the valve train $F_{\text{C4}}$ and centrifugal force $F_{\text{M}}$ act on the engine block and the rotating shafts simultaneously. Figure 2 shows the waveforms of exciting forces at no load (idling) condition in which combustion pressure and fuel injection pressure are measured and others are calculated results.
2.3 Analytical model of rotating shaft and bearings

Vibration characteristics of crankshaft, injection pump shaft and valve train shaft are taken into consideration in this analysis. Shafts of injection pump and valve train are approximated as isotropic axis system. As vibration mode shape of crankshaft is coupled of bending, torsional and axial deformation, crankshaft is treated as anisotropic axis system. Shaft length of intermediate gear is short enough to be approximated as rigid shaft whose vibration components are rotation, rocking and translation. Therefore, shear force and bending moment at bearing induce vibration of front side of engine block.

The dynamic properties at bearing are approximated by stiffness and damping which are series connection of oil film stiffness and local structural stiffness. The locus of eccentricity of the shaft center is determined by bearing force that changes with the crank angle and the stiffness and damping of oil film are evaluated at each crank angle. The dynamic stiffness at bearing is considered only for translation displacement and moment stiffness is neglected in this study. Detailed calculation procedure was presented in (Ohta, et al., 1994). As the helical gear generates the radial force and axial force, the crank shaft and gear train shafts are connected to the engine block by the radial bearing and thrust bearing. In the case of this crankshaft, one thrust bearing is installed to the flywheel side.

2.4 Analytical model of gear meshing and impact

Figure 3 shows the gear meshing force and impact model (Ohta, et al., 2002, Theodossiades, et al, 2007). In the gear train shown in Fig.3(a), torque and force are transmitted by the dynamic stiffness at gear meshing point. In the case of the helical gear with helix angle \( \beta \), the dynamic tooth load \( f_{12} \), which is normal to the tooth surface, has the radial and axial components of the shaft system as shown in Fig.3(a) and (b) (Cai, 1995). Therefore, gear train shaft is connected to the engine block by the radial bearings and one thrust bearing. When the steady torque is large enough not to cause separation of gear tooth, stiffness of meshing point changes with the torque and contact point of the gear tooth (Dion, et al, 2009). But in the case of idling operation discussed in this paper, separation and impact of gear tooth are generated by the rotational fluctuation of crankshaft and impulsive torque due to fuel injection.

![Diagram of gear meshing and impact](image_url)
Fig.3(c) and (d) show the analytical model of gear meshing which is approximated by the spring K and viscous damping C that corresponds to the stiffness of the gear tooth and damping of lubricant and each meshing point has a clearance $\delta_2$ so called backlash. In this impact model, it is assumed that oil film of lubricant, which thickness is $\delta_2 - \delta_1$, has the damping effect before metal contact of gear tooth and gear meshing spring begins to generate the restoring force when oil film stiffness approaches to the surface roughness.

3. Equation of motion

(1) Engine block

Equation of motion of engine block is written in the stationary coordinate as follows:

$$M_0 \ddot{x}_0 + C_0 \dot{x}_0 + K_0 x_0 = -F_{B1} - F_{B2} - F_{B3} - F_{B4} + F_{G0} + F_{E0} - F_{C3} - F_{C4}$$ (1)

Bearing force caused by rotating shaft is expressed by use of stiffness matrix $k_B$, damping matrix $c_B$ at the connecting point

$$F'_{Bq} = \begin{bmatrix} \vdots \\ f'_{Bq} \\ \vdots \end{bmatrix} = -F_{Bq}, \quad q = 1 \sim 4, \quad f'_{Bq} = -k_{Bq}(x_{B0} - x_{Bq}) - c_{Bq}(\dot{x}_{B0} - \dot{x}_{Bq})$$ (2)

(2) Crankshaft

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.

$$M_1 \dddot{u}_1 + (C_1 + \Delta C_1) \ddot{u}_1 + (K_1 + \Delta K_1)u_1 = F_M + F_c + T^t(F_{B1} + F_{G1})$$ (3)

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 $u_1$ is determined in the rotating coordinate. External forces to be considered are the reaction force of the main bearing impact $F_{B1}$ and meshing 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 transformation matrix from the rotating coordinate to the stationary coordinate. Gear meshing force $T^tF_{G1}$ defined in the rotating coordinate is given by the following equations;

$$T^tF_{G1} = \begin{bmatrix} 0 \\ \vdots \\ T^tF_{12} \\ \vdots \\ 0 \end{bmatrix} = -K_{11}u_1 - C_{11}\dot{u}_1 + K_{12}x_2 + C_{12}\dot{x}_2$$ (4)

where $u_1$ and $x_2$ are vibration displacement of crank gear and intermediate gear at meshing point and those values are evaluated by use of the translation and rotational displacement of the axis center. It is assumed that intermediate gear shaft has no elastic vibration mode below 4kHz in which frequency components of engine noise are dominant.

(3) Intermediate gear shaft

Equation of motion of intermediate gear shaft in the stationary coordinate is

$$M_2 \dddot{x}_2 + C_2 \dddot{x}_2 + K_2 x_2 = F_{G2} + F_{B2}$$ (5)

where $F_{G2}$ is gear meshing force and $F_{B2}$ is reaction of bearing force. As intermediate gear has three meshing points with crankshaft, fuel injection pump and valve train, meshing force of intermediate gear $F_{G2}$ is given by

$$F_{G2} = -K_{22}x_2 + K_{21}u_1 + K_{23}x_3 + K_{24}x_4 - C_{22}\dot{x}_2 + C_{21}\dot{u}_1 + C_{23}\dot{x}_3 + C_{24}\dot{x}_4$$ (6)

(4) Fuel injection pump shaft

Equation of motion of injection pump shaft in the stationary coordinate is described by the following equation considering the gear meshing force $F_{G3}$, reaction of bearing force $F_{B3}$ and force and moment due to injection pressure $F_{C3}$.

$$M_3 \dddot{x}_3 + C_3 \dddot{x}_3 + K_3 x_3 = F_{G3} + F_{C3} + F_{B3}$$ (7)
Gear meshing force with intermediate gear $F_{G3}$ is

$$F_{G3} = -K_{G3}^G x_3 + K_{G2}^G x_2 - C_{G3}^G \ddot{x}_3 + C_{G2}^G \ddot{x}_2 \quad (8)$$

(5) **Valve train shaft**

Equilibrium equation of valve train shaft considering gear meshing force $F_{G4}$, reaction force of bearing load $F_{B4}$ and valve train induced force and moment $F_{C4}'$ is written in the stationary coordinate.

$$M_4 \ddot{x}_4 + C_4 \dot{x}_4 + K_4 x_4 = F_{G4} + F_{C4}' + F_{B4} \quad (9)$$

Gear meshing force $F_{G4}$ is

$$F_{G4} = -K_{G4}^G x_4 + K_{G2}^G x_2 - C_{G4}^G \ddot{x}_4 + C_{G2}^G \ddot{x}_2 \quad (10)$$

(6) **Equation of motion of the total engine system**

Combining Eq.(1)~Eq.(10) yields equation of motion of the total engine system:

$$
\begin{bmatrix}
M_0 & M_1 & M_2 & M_3 & M_4 \\
0 & M_5 & M_6 & M_7 & M_8 \\
C_0 + C_{00}^B + C_E & -C_{01}^B & -C_{02}^B & -C_{03}^B & -C_{04}^B & \ddot{x}_0 \\
-C_{10}^B & C_1 + \Delta C_1 + C_{11}^B + C_{11}^G & -C_{12}^B & 0 & 0 & \ddot{u}_1 \\
-C_{20}^B & -C_{21}^G & C_2 + C_{22}^B + C_{22}^G & -C_{23}^G & -C_{24}^G & \ddot{x}_2 \\
-C_{30}^B & 0 & -C_{32}^G & C_3 + C_{33}^B + C_{33}^G & 0 & \ddot{x}_3 \\
-C_{40}^B & 0 & -C_{42}^G & 0 & C_4 + C_{44}^B + C_{44}^G & \ddot{x}_4
\end{bmatrix}
\begin{bmatrix}
x_0 \\
\dot{x}_0 \\
\dot{x}_1 \\
\dot{x}_2 \\
\dot{x}_3 \\
\dot{x}_4
\end{bmatrix}

= \begin{bmatrix}
F_{S0} + F_{G0} - F_{C3} - F_{C4} \\
F_C + F_M \\
F_{C3} \\
F_{C4} \\
0 \\
F_{C4}
\end{bmatrix}
\quad (11)

In the above equation, $C_{rq}^B$, $K_{rq}^B$ are the damping and stiffness matrix of bearing between structure $r$ and structure $q$. $C_E$, $K_E$ are the damping and stiffness matrix of engine mount.

3.2 Vibratory characteristics of engine components

Vibration characteristics of engine component such as engine block, crankshaft and gear train are determined with free boundary condition. Eigen-value equation of engine component $q$ is

$$(-\omega^2 M_q + K_q) x_q = 0 \quad (12)$$

Solving eigen-value equation yields natural frequency $\omega_{qn}$, mode shape $\phi_{qn}$ and modal mass $m_{qn}$ of the $n$-th mode.

$$m_{qn} = \phi_{qn}^T M_q \phi_{qn} \quad (13)$$
Orthogonality conditions of mass matrix $M_q$ and stiffness matrix $K_q$ are
$$\begin{align*}
\tilde{M}_q &= \Phi^t_{q_1} M_q \Phi_{q_1} = \begin{bmatrix} \tilde{m}_{q_1} \\ \vdots \\ \tilde{m}_{q_N} \end{bmatrix}, \\
\tilde{K}_q &= \Phi^t_{q_1} K_q \Phi_{q_1} = \begin{bmatrix} \tilde{m}_{q_1} \omega_{q_1}^2 \\ \vdots \\ \tilde{m}_{q_N} \omega_{q_N}^2 \end{bmatrix}
\end{align*}$$

Orthogonality condition on damping matrix is assumed:
$$\begin{align*}
\tilde{C}_q &= \Phi^t_{q_1} C_q \Phi_{q_1} = \begin{bmatrix} \tilde{c}_{q_1} \\ \vdots \\ \tilde{c}_{q_N} \end{bmatrix}, \\
\tilde{K}_q &= \Phi^t_{q_1} K_q \Phi_{q_1} = \begin{bmatrix} \tilde{m}_{q_1} \omega_{q_1} \\ \vdots \\ \tilde{m}_{q_N} \omega_{q_N} \end{bmatrix}
\end{align*}$$

### 3.3 Eigenmode expansion

Vibration displacement $u_1$ of the rotating crankshaft observed in the rotating coordinate is expressed by the linear combination of the eigenmode $\Phi_{1n}$ which are determined in the stationary coordinate. Vibration displacement of the engine block and gear train system $x_q$ ($q=0,2,3,4$) are also expressed by the eigenmode $\Phi_{qn}$ determined in the stationary coordinate.

$$u_1 = \sum \Phi_{1n} a_{1n} = \bar{\Phi}_1 \bar{a}_1, \\
x_q = \sum \Phi_{qn} a_{qn} = \bar{\Phi}_q \bar{a}_q, \quad q = 0,2,\ldots,4$$

where $a_q = (a_{q1}, a_{q2}, \ldots, a_{qN})^t$ is the modal response vector and $\Phi_q = [\Phi_{q1}, \Phi_{q2}, \ldots, \Phi_{qN}]$ is the eigen mode shape matrix of the structure $q$. Substituting Eq.(16) into Eq.(11) and multiplying $\Phi^t_q$ from the left side, one can rewrite the equation of motion in the modal coordinate as follows:

$$\ddot{\bar{M}} \bar{a} + \ddot{\bar{C}} \bar{a} + \ddot{\bar{K}} \bar{a} = \bar{F}$$

where $\bar{a} = (a_0, a_1, a_2, a_3, a_4)^t$

Substituting the external force like combustion pressure and piston slap force, modal response $\bar{a}$ are calculated by the time integral technique. Vibration response of each structure are obtained by Eq.(16) and bearing force and gear meshing forces are calculated by Eq.(2),(4),(6),(8),(10).

### 4. Prediction of Engine Noise

#### 4.1 BEM formulation

Governing equation of acoustic field when the engine block surface vibrates at angular frequency $\omega$ is give by

$$\nabla^2 P + k^2 P = 0, \quad k = \frac{\omega}{c}$$

Using Green’s theorem, acoustic pressure of the point $l$ on the boundary surface is written by

$$P_l = -\frac{1}{2\pi} \int_\gamma \left( \frac{\partial P}{\partial n} - P \frac{\partial \phi}{\partial n} \right) dS, \quad \phi = \frac{e^{-jkr}}{r}$$

When boundary of the acoustic field is divided into small elements and acoustic pressure and pressure gradient are assumed to be constant on each small element, acoustic pressure $P_m$ in the boundary element $m$ is expressed by the following equation (Nakamura, et al., 1990).

$$\sum_{m=1}^{N} h_m P_m = \sum_{m=1}^{N} g_m \frac{\partial P_m}{\partial n}, \quad \bar{H}P = \bar{G} \frac{\partial P}{\partial n}$$

Relation of vibration acceleration on the surface $A_n$ and acoustic pressure $P$ is

$$\frac{\partial P}{\partial n} = -P A_n$$

If vibration acceleration $A_n$ on the engine block surface is given, acoustic pressure distribution $P$ on the surface can be determined by solving the Eq.(20). Matrix element of $H$ and $G$ are

$$h_m = \begin{bmatrix} -1 \\ \frac{1}{2\pi} \int_\gamma \left( \frac{e^{-jk_s}}{r_m} \right) dS_m \\ \vdots \\ \frac{1}{2\pi} \int_\gamma \left( \frac{e^{-jk_l}}{r_m} \right) dS_m \end{bmatrix}, \quad g_m = \frac{1}{2\pi} \int_\gamma \frac{e^{-jk_m}}{r_m} dS_m$$

4.2 Acoustic radiation power

If vibration acceleration $A_n$ on the engine block surface is given, acoustic pressure distribution $P$ on the surface can be determined by solving the Eq.(20). Acoustic Power $W(\omega)$ radiated from engine block can be represented by

$$W(\omega) = -\frac{1}{2j\rho\omega} \int_S P \cdot \frac{\partial P}{\partial n} dS$$  \hspace{1cm} (23)

5. Effect of the torsional vibration of crankshaft

5.1 Calculating condition and torsional vibration of crankshaft

Present method is applied to 4 cycle - 4 cylinders high speed diesel engine with displacement of 3.5 liter. Backlash of the gear train is assumed as ±40μm. Engine rotational speed changes from $N_E=500$ rpm to $N_E=4000$ rpm in the idling condition.

Figure 4 shows the calculated results of angular acceleration $\dot{\theta}_1$ of the crank gear at low and high engine rotational speed. At the low engine speed $N_E=700$ rpm, one may see the periodic amplitude of rotational fluctuation synchronizing with the combustion of each cylinder and crankshaft and gear train vibrate in a rigid body motion. On the other hand, at the high engine rotational speed $N_E=2500$ rpm, rigid body motion of crankshaft becomes small and torsional vibration remarkably appears. Large amplitude of crankshaft vibration is generated just after combustion TDC(Top Dead Center) of #1 and #2 cylinder, because crankshaft has the torsional eigen mode at the frequency of 400Hz in which torsional amplitudes of #1 and #2 crankpin are larger than those of #3 and #4 crankpin. Figure 5 shows the comparison of the gear impact force $F_{12}$ between the crank gear and intermediate gear at low and high engine speed.

![Fig.4 Angular acceleration of crank gear](image1)

![Fig.5 Gear meshing force (crankshaft vs. intermediate gear)](image2)
At the low engine speed, gear tooth impact at the positive side (drive side) and negative side are seen with synchronization of the angular acceleration of crank gear. At the high engine speed, large gear tooth impact are caused just after the combustion top dead center of the #1 and #2 cylinder and these timings coincide with the crank angle when the rotational acceleration of the crank gear reaches the maximum value. This implies that severe gear tooth impact is induced by the torsional vibration of the crankshaft.

5.2. Estimation of the natural frequency of the crankshaft torsional vibration

Figure 6 shows the calculated results of the rotational acceleration of crank gear versus engine speed, represented by engine rotational order of 6th - 10th component. Clear resonance peaks are observed in the 7th - 10th order components. As the resonance peak appears at the rotational speed $N_E=3400$rpm in the 7th order component, the resonance frequency is calculated to be $3400/60\times7=397$Hz. The other order components show the similar results around 400Hz. This resonance frequency corresponds to the first torsional vibration mode of the crankshaft shown in Fig.7. In this torsional vibration mode, rotational amplitude at the pulley side(#1 cylinder side) is larger than that at the flywheel side because the rotary inertia of the flywheel is much greater than pulley. Thus, this torsional vibration mode is more excited by the forces acting on the #1 and #2 crank pin and this feature corresponds to the waveform shown in Fig.4, in which large amplitude of crankshaft vibration is generated just after combustion TDC of #1 and #2 cylinder compared with #3 and #4 combustion TDC at the high engine rotational speed $N_E=2500$rpm.

![Fig.6 Higher order frequency components of rotational acceleration](image)

![Fig.7 Torsional mode shape of crankshaft](image)
5.3 Change of location of gear train and engine noise reduction

As the first torsional mode of the crankshaft is stimulated with the increasing of the engine speed, the gear tooth impact between crank gear and intermediate gear is intensified. Since the rotational displacement at the pulley side is larger than the flywheel side in this torsional mode, the change of the location of gear train from pulley side to the flywheel side as shown in Fig.8 is expected to reduce the gear tooth impact and its related engine noise. In this section, the effect of the change of the location of gear train is evaluated by the numerical analysis mentioned in section 2 and 3. Figure 9 shows the change of the waveform of the rotational acceleration of the crank gear after the location of the gear train is changed from the pulley side to the flywheel side. Before the change of the gear train location, large fluctuation of the rotational acceleration of the crank gear is seen just after the combustion TDC of the #1 and #2 cylinder. These rotational acceleration is lessened to one fourth after the change of gear train location. Amplitudes of the rotational acceleration just after the combustion of each cylinder are the almost same.

![Fig.8. Change of location of gear train](image)

![Fig.9 Angular acceleration of crank gear](image)
Figure 10 shows the change of the waveform of gear tooth impact force. After the change of the gear train location, severe gear tooth impact after the combustion TDC of the #1 and #2 cylinder diminished. This shows that reduction of the torsional vibration can lessen the gear tooth impact in the gear train. Figure 11 shows the frequency spectrum of the gear meshing force between crank shaft and intermediate gear. As shown in Fig.10, peak values of the gear impact force just after the combustion TDC of the #1 and #2 cylinder are reduced by suppressing the crankshaft torsional vibration of which frequency is around 400Hz. Since gear impact force is a pulse-like waveform which has high frequency components, reduction of the gear impact force above the frequency of 400Hz is seen in the frequency spectrum of Fig.11.

Figure 12 shows the change of the engine noise spectrum, 1m apart from the engine block, in 1/3 octave band frequency after the change of the gear train location. Engine noise spectrum above the frequency of 800Hz decreased due to the suppression of the gear impact which has the high frequency components shown in Fig.11. Reduction of the overall noise level is estimated by 2.2dB.
6. Conclusion

As the small-sized high speed diesel engine used for construction machinery and generator is operated under continuous heavy load condition, a gear train is employed to drive the fuel injection pump and valve train. When a loading torque is small at the idling condition, the gear tooth separation and impact are caused by the fluctuating torque and they increase the engine noise level. Particularly when the torsional vibration of the crankshaft occurs and its rotational amplitude is magnified at a specified rotational speed, the gear impact is intensified and it leads to increase of engine noise. Author has developed the theoretical procedure using modal analysis technique 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. This method is applied to examine the effect of torsional vibration of crankshaft on the gear impact force and radiated noise of the actual engine. It is revealed that the first torsional vibration mode, which frequency is about 400Hz, is stimulated with increasing of the engine rotational speed and engine noise due to the gear tooth impact increases. Based on this analysis, noise reduction measure is offered, which is the change of the location of the gear train from the pulley side to the flywheel side of the crankshaft. This measure is expected to reduce the generation of the first torsional mode of the crankshaft and to abate the engine noise level.

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