Reduction of Diesel Knock Noise by Controlling Piston Vibration Characteristics

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ABSTRACT: It has been required recently that diesel engines for passenger cars should meet various requirements, such as low noise, low fuel consumption, low exhaust gas emission and high power. The key to improve the noise is to reduce a combustion noise known as diesel knock noise. An experiment of the noise transfer path from a piston through a connecting rod and a crankshaft to an engine during engine operation revealed that piston resonance was a noise source. To absorb piston resonant energies, new piston structure with Natural Sound Smoother which resonates in the opposite phase to the piston is designed.

KEY WORDS: vibration, noise, ride comfort, power train, test and analysis technology, Diesel knock noise, Dynamic Damper [B3]

1. Introduction

During diffusive combustion of a direct-injection diesel engine, fuel is injected from the injector when the combustion chamber is ready for ignition and cylinder pressure rapidly rises to trigger self-ignition at the several areas. Since high thermal efficiency can be obtained by fast burn, Premixed charge compression ignition and Homogeneous-Charge Compression Ignition will be employed in the future⁽¹⁻³⁾. However, from Noise and Vibration point of view, due to rapid cylinder pressure applied to engine components by fast burn, engine vibration occurs and “rattling” intermittent diesel knock noise is generated. To achieve both ultimate thermal efficiency and reduction in diesel knock noise, a new engine structure needs to be developed to reduce diesel knock noise without depending on combustion control.

It is known that the dominant frequency of diesel knock noise exists near the 3.15 kHz one third octave band⁽⁴⁾. Therefore, by focusing on the 3.15 kHz one third octave band, the mechanism of diesel knock noise was identified and a new low vibration engine structure was developed.

2. Characteristics of Diesel Knock Noise

Inline 4 cylinder 2.2L direct injection diesel engine was used for the test. To understand the frequency characteristic of diesel knock noise better, the fuel quantity of pre-injection was reduced and combustion excitation force was increased intentionally to generate distinct diesel knock noise. Figure 1 shows the spectrogram of near field radiated engine noise measured at 2000rpm (900 kPa Brake Mean Effective Pressure). Notable noise peak exists at 3.5 kHz.

Fig. 1 Characteristics of Diesel Knock Noise analyzed by short-time Fourier transform (genesis timing of combustion and engine noise)

3. Diesel Knock Noise Mechanism

The influence of noise transfer path in the internal system was analyzed⁽⁵⁾. Figure 2 shows the path which transfers excitation force from piston through connecting rod and crankshaft to engine block. During combustion, oil acts as a spring and damper in elastic hydrodynamic lubrication and transfers force to the engine internal parts. Since the conditions of this spring and damper vary with engine operating conditions and crank angle, it is difficult to replicate the boundary conditions of oil under static hammering test. Therefore, internal vibration behavior was directly measured during actual engine operation.

Fig. 2 Noise transfer path

Figure 3 and Figure 4 show the location of an accelerometer on the piston and connecting rod. Figure 5 shows the procedure to retrieve signal from reciprocating parts using a link arm device. The accelerometer cable is bonded to the link arm so that the cable can be routed to the outside the engine. Accelerometers with built-in amplifier are used to exclude the electrical resistance variation caused by the cable twist during the operation of the link arm.

Fig. 4 Vibration measurement of connecting rod

Fig. 3 Vibration measurement of piston during engine operation

Figure 6 shows time-history graphs measured during combustion in No. 4 cylinder. The moment self-ignition occurs in the cylinder, cylinder pressure level rapidly rises and vibrates the piston first. With delay, vibration is transferred to the connecting rod, crankshaft and finally to the engine block. Therefore, it can be concluded that combustion excitation force is transferred from the internal system to the engine block.

Fig. 5 Diagram of link arm

Fig. 6 Measured cylinder pressure, vibration during combustion

Figure 7 shows spectrum of cylinder pressure and internal component vibration during combustion. The peak of cylinder pressure is not present between 3 kHz and 4 kHz. However, significant vibration peaks of the piston and other internal components exist between 3 kHz and 4 kHz. The actual measured operational deflection shape at 3.5 kHz is shown in Figure 8. Connecting rod contracts and extends like a spring at 3.5 kHz.

Fig. 7 Spectrum of measured cylinder pressure and vibration

Fig. 8 Measured 3.5kHz operational deflection shape of piston and connecting rod with crank angle in Fig.6

Therefore, it can be presumed that a single degree of freedom system consisting of piston, piston pin, and connecting rod small end as piston mass, connecting rod as a spring and crankshaft as a fixed end as shown in Figure 9 is the cause of the vibration peak between 3 kHz and 4 kHz.

Fig. 9 Predicted single degree of freedom resonance model

Figure 10 shows the calculated relationship between piston mass, connecting rod stiffness and the piston resonance frequency. The resonance frequency of 2.2 L engine exists between 3 kHz and 4 kHz. The resonance frequency of above 1.5L engine with less piston mass and the spring constant of connecting rod also exists between 3 kHz and 4 kHz. As long as connecting rod stiffness is proportional to piston mass, resonance frequency always appears between 3 kHz and 4 kHz even with different engine displacements⁽⁶⁾. In order to confirm the actual piston resonance frequency both of 1.5L Diesel engine and 2.0L Gasoline engine, piston vibration during combustion was measured experimentally. Figure 11 shows the 1.5L Diesel engine piston has resonance at 3.5kHz. Vibration of 2.0L Gasoline engine was measured by advancing spark ignition intentionally to cause rapid cylinder

pressure increase. Figure 12 shows the piston resonance of 2.0L Gasoline engine detected at 3.3kHz.

Fig. 10 Mass/ spring constant vs. piston resonance frequency

Fig. 11 Spectrum of the measured piston vibration of 1.5L DE

Fig. 12 Spectrum of the measured piston vibration of 2.0L GE

4. Diesel Knock Noise Reduction

4.1. Reduction by Changing Engine Specification

To reduce the resonance of piston/ connecting rod generated between 3 kHz and 4 kHz, internal structure modification was studied. First, reduction in the force transferred to crankshaft was analyzed by changing piston mass and spring constant of connecting rod. Since significant improvement in material damping rate is unrealistic, it is not included in this study. Figure 13 shows FEM model used for the analysis. Piston, piston-pin and connecting rod small end are all connected rigidly. At big end, bearing and center fixing end are connected together with oil film acting as a spring and oil film stiffness is tuned to match with the measured resonant frequency of piston/ connecting rod.

Fig. 13 FEM model used for the analysis

Figure 14 shows the prediction results of change in force transferred to the crankshaft by varying piston mass and connecting rod stiffness. As piston mass is reduced, the transferred force is also reduced. However, significant reduction in piston mass is unrealistic. Even with 10% reduction in mass, the transferred force can be reduced only by 0.4dB.

The transferred force can be also reduced by reducing connecting rod stiffness. However, since max cylinder pressure of diesel engine is high, the significant reduction of connecting rod stiffness is also unrealistic because of the reliability requirements. Even with 10% reduction in stiffness, the force can be reduced only by 0.4dB. Therefore, it can be concluded that reduction in transferred force to crankshaft by changing engine specifications is not effective.

Fig. 14(a) Influence of piston mass

Fig. 14(b) Influence of connecting rod stiffness

4.2. Reduction by Natural Sound Smoother

To reduce high frequency annoying clatter and to improve sound quality, Natural Sound Smoother has been devised, which aggressively controls piston vibration characteristics like a dynamic vibration reducer. Figure 15 shows the diagram of Natural Sound Smoother that resonates in the opposite phase to the piston and absorbs piston vibration energy to reduce the transferred force.

Fig. 15 Piston vibration damping mechanism

Figure 16 shows the predicted reduction in transferred force with 2 types of Natural Sound Smoother weighing 1% and 5% of piston mass. Since significant reduction in peak resonance is seen, significant vibration reduction can be expected. As the mass of the Natural Sound Smoother is increased, the transferred force is reduced.

Fig. 16 Effect of Natural Sound Smoother

Therefore, to reduce diesel knock noise, the mass of Natural Sound Smoother should be increased. However, due to installation space, thermal damage and layout requirements, the size of Natural Sound Smoother is limited. Considering all requirements, Natural Sound Smoother weighing 2.9 to 8.7% of piston mass was designed and installed in the piston pin. Figure 17 shows the dimension, operation mode and assembly drawing of the Natural Sound Smoother⁽⁷⁾. Since the piston resonance exists at 3.5 kHz, the operating frequency of the Natural Sound Smoother must be set at 3.5 kHz. The operating frequency was tuned to 3.5 kHz with spring property. To meet stress requirements, the diameter and length of the spring were optimized.

Each mass on the right and left side resonates vertically in the same phase

Fig. 17 Structure and operation mode of Natural Sound Smoother

Reduction in resonance and internal vibration and diesel knock noise at 3.5 kHz were verified by installing the Natural Sound Smoother in the engine. Crank pin journal vibration, engine block vibration and near field engine noise with or without the Natural Sound Smoother were compared. Figure 18 shows the effect of the Natural Sound Smoother weighing 2.9% of piston mass. Crank pin vibration is reduced at 3.5 kHz, therefore, the force transferred to the crankshaft is reduced as predicted. Due to the vibration reduction in the internal components, engine block vibration and near field engine noise are also reduced.

To verify vibration reduction by increasing Natural Sound Smoother mass, measurement with a Natural Sound Smoother weighing 8.7% of piston mass was also conducted. Figure 19 shows the results. Further reduction in vibration of crank pin, engine block and near field engine noise can be seen at 3.5 kHz. Absolute value of the amplitude is reduced and the range of effective Natural Sound Smoother operating frequency is increased from 260Hz to 460Hz, therefore, it indicates significant reduction in noise energy. Furthermore, in auditory evaluation, it was confirmed noise fluctuation was reduced and sound quality was improved. We should consider if installing Natural Sound Smoother makes low-frequency vibration worse due to a gain of 2nd harmonic engine vibration by increasing reciprocation inertial mass. Installing Natural Sound Smoother weighing 8.7% of piston mass equals to 8.7% increase of reciprocation inertial mass. This means only 0.72dB increase of vibration, so that there is few difference of low-frequency vibration between with and without Natural Sound Smoother. Thus, reduction of diesel knock noise by Natural Sound Smoother is achieved with little increase of low-frequency vibration.

The above results verify that diesel knock noise can be reduced by modifying the engine internal structure such as piston and connecting rod from Noise point of view.

Fig. 18 The actual effect of Natural Sound Smoother weighing 2.9% [$1\text{m/s}^2 = 100\text{dB}$]

Fig. 19 The actual effect of Natural Sound Smoother weighing 8.7% [$1\text{m/s}^2 = 100\text{dB}$]

5. Conclusion

The followings are the conclusions of the noise peak between 3 kHz and 4 kHz which characterizes diesel knock noise.

1. The vibration measurement of piston, connecting rod and crankshaft during combustion demonstrates that the vigorous vibration generated is caused by piston vibration
2. The vibration mechanism of piston is the resonance of the single degree of freedom system consisting of the piston as a mass and the connecting rod as a spring and the crankshaft as a fixed end.
3. Since piston mass must have proportional relationship with the connecting rod stiffness due to reliability requirements, engines with different displacements also have resonance between 3kHz and 4kHz.
4. Tests have proven if piston vibration energy is absorbed by Natural Sound Smoother that resonates in the opposite phase to the piston, diesel knock noise can be reduced.
5. As Natural Sound Smoother mass per piston mass is increased, diesel knock noise can be reduced. The lighter the piston mass is, the larger the damping effect can be.

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