Noise Sources of Air Cooled Motor Cycle Engines

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Abstract

A frequency analysis technique for each time segment of the vibration and noise responses of a running engine is proposed to analyze the contribution of each response to the total. The time histories of the one—third octave band frequency components of engine noise are also employed to identify the generation time of each noise response and its frequency characteristic. An approach for diagnosis of impact origins is carried out through vibration isolation and elimination of engine parts. Using these analysis techniques, the noise and vibration sources in a small air—cooled two—stroke cycle engine are clarified. As a result of these investigations, it was possible to obtain information about the generation times and frequency characteristics of radiated noise corresponding to each impact at any particular moment in a running engine. It was also ascertained that the impact of the primary reduction gears was the predominant cause of noise in no load operation of the test engine.

Introduction

Impacts occurring in various parts of an engine give rise to vibration, which is then transmitted through the structures and radiates as noise from engine surfaces. These impacts may be caused by combustion, piston slaps, backlash of gears, clearance between moving components, etc. Therefore, engine noise and its vibration throughout each cycle are to be regarded as a set of responses corresponding to each impulsive force.

While engine noise and vibration have usually been measured in terms of the time averages such as noise levels and vibration levels or their band frequency components, in clarifying the causes of noise using such measurements, the noise radiation is implicitly assumed to be steady. Hence we need another method to clarify the generation of noise and its causes in engines.

In this paper, the frequency analysis for each time segment involved in one engine cycle of vibration and noise is examined. The time histories of each band frequency component of engine noise are investigated to clarify the frequency characteristic and generation time of response corresponding to each impact which is caused at various points within the engine structure. An attempt is also made to identify the origin of the impact by isolating vibration and/or eliminating the engine parts.

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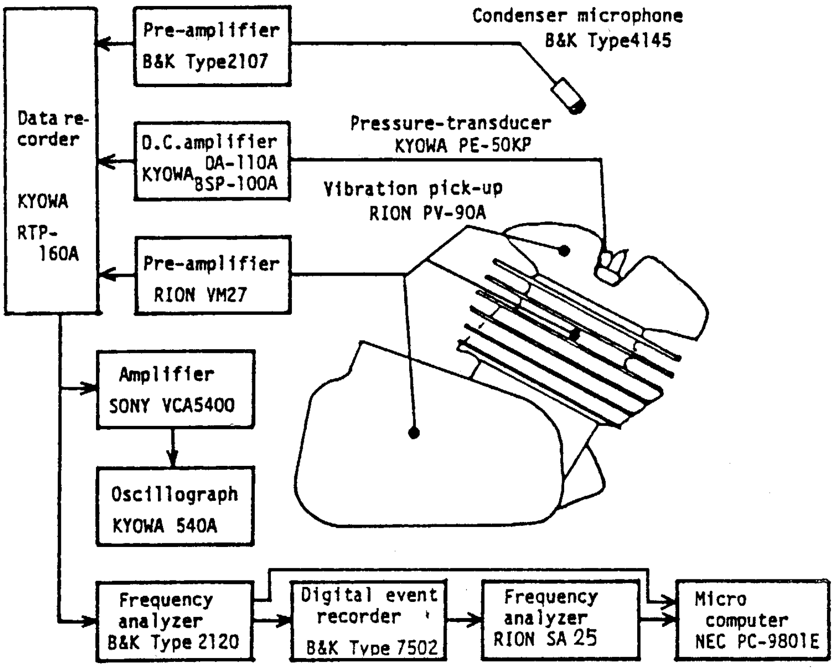


Fig. 1 Block diagram of measuring and analyzing system.

Experimental Procedures

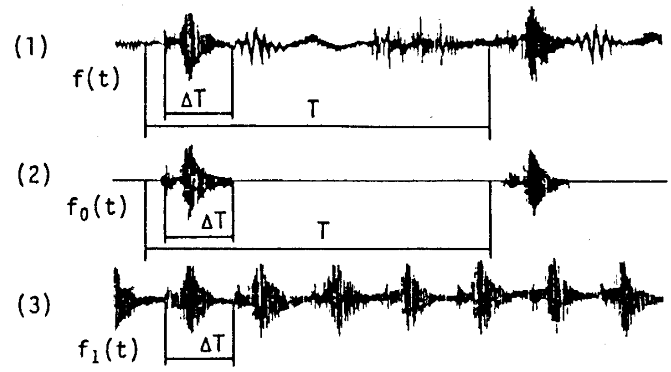
A block diagram of the measuring and analyzing system is shown in Fig.1. The test engine was an air—cooled, two—stroke, single cylinder S. I. engine (125cm³, 9.5kW/7000rpm), and was set in an anechoic room. The intake and exhaust pipes were lagged with sound absorbing material and lead sheets to insulate the noise radiated from their surfaces.

The engine noise was detected by a condenser microphone (B&K4145) located over the cylinder head of the engine at a distance of 0.5m. Piezo type acceleration transducers (Rion PV90A) were set on the middle of the cylinder head, the cylinder block and the crank case. A strain gauge type pressure transducer (Kyowa KE50KP) was mounted flush with the inner wall of the combustion chamber to track the progress of cylinder pressure. An impulse hammer (PCB 230A03) was employed to investigate the vibration isolating effects of a rubber sheet inserted between the cylinder block and the crank case.

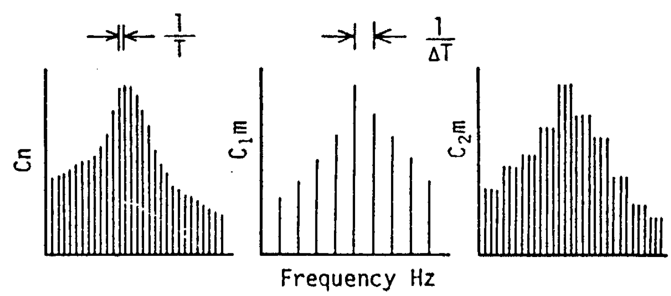
Each signal in a time segment was sampled by a digital event recorder (B&K 7502) synchronized with the crank marks of the engine, and analyzed by a one—third octave band frequency analyzer (Rion SA21,SA10). The time history of each band frequency component of the signals was observed through a band pass filter (B&K 2120). These signals were processed by a micro—computer (NEC PC9801) through an A/D converter.

Procedure of Frequency Analysis for Each Response

A frequency analysis of the signal having a duration ΔT among the group of



(a) Correction level L_1 $(20 \log \frac{\Delta T}{T})$



(b) Correction level L_2 $(10 \log \frac{T}{\Delta T})$

Fig. 2 Frequency analysis for each response signal.

responses within the period T is performed to obtain the frequency characteristic of each response signal which occurs in a particular time segment in one cycle of engine running as shown in Fig. 2 (a)−(1).

By regarding the instantaneous signal ΔT as a part of the signal f(t), we can directly obtain the frequency characteristic using a digital computer. But using a real time analyzer, it is difficult to analyze a signal of this sort because of its response time. Then for the signal $f_1(t)$ repeated with the period ΔT, the correction level L_1 ($L_1 = 20\log(\Delta T/T)$) was necessary due to changes in the repeated cycle.

In regards to the band pass filter analysis, a correction for differences in the number of the line spectra contained in each band is also required. It is assumed that the levels of line spectra of f_0 (t) are kept constant within the frequency interval of $1/\Delta T$ as shown in Fig. 2 (b). Then, we obtain the correction level L_2 ($L_2 = 10\log(T/\Delta T)$). So, the overall correction levels are expressed by $L_1 + L_2$ ($L_1 + L_2 = 10\log(\Delta T/T)$) in decibels.

Fig. 3 shows the results of frequency analysis through these procedures for each time segment from ⓘ to ⓘⓥ (shown in upper part) in one cycle of engine noise. As is evident from this figure, the engine noise radiated in each moment has different frequency characteristics.

To confirm the validity of these procedures, the sum of the results ⓘ through ⓘⓥ was

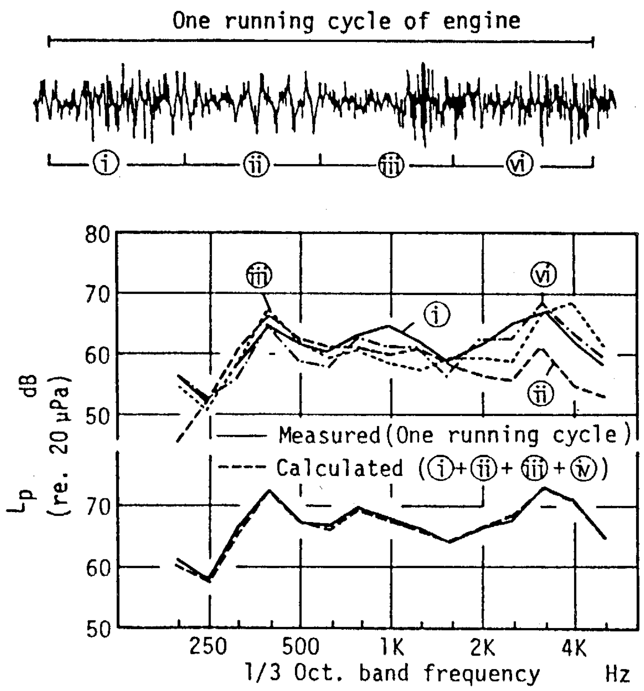


Fig. 3 Frequency characteristics of engine noise radiated in each time segment.

compared to the measurement for one entire cycle. A fairly good agreement is obtained between them.

This procedure makes it possible to clarify the frequency characteristic of $f_0(t)$ by analyzing $f_1(t)$ with the help of the correction levels L_1 and L_2.

Vibration and Noise in Idling Operation

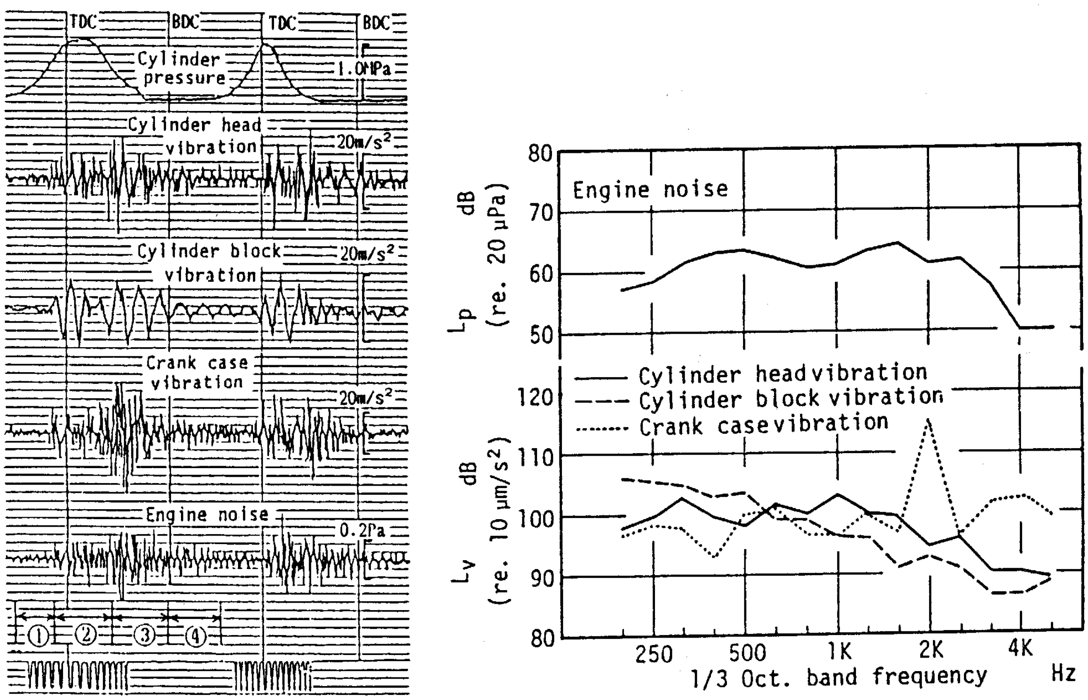
Time Histories of Engine Noise and Vibration.

The noise and vibration acceleration time traces in an idling engine are shown in Fig. 4. In the idle of this engine, three non−combustion cycles follow each combustion cycle. The combustion cycle is shown on the left and the non−combustion cycle is shown on the right in Fig.4(a). From this figure, it seems that a strong impact affects the engine structure around the time that the exhaust port is open in the expansion stroke.

The frequency characteristics of the engine noise and vibration accelerations are shown in Fig. 4 (b). Although keen vibration peaks are found at the crank case of 2kHz and 4kHz, there is no evident peak in noise at these frequencies. Now we need another means to clarify the relationship between the vibrational response and engine noise in running operation.

Frequency Analysis for Each Time Segment.

Fig. 5 shows the results of the frequency analysis for each time segment of engine noise and vibration. Each time segment corresponds to the indication shown in Fig. 4 (a), i. e., ① compression period, ② ignition and expansion period, ③ first half of the



(a) Time histories of vibration and noise

(b) Frequency characteristics of noise and vibration of engine

Fig. 4 Responses of vibration and noise, and their frequency characteristics in idling engine, (1500rpm, no load).

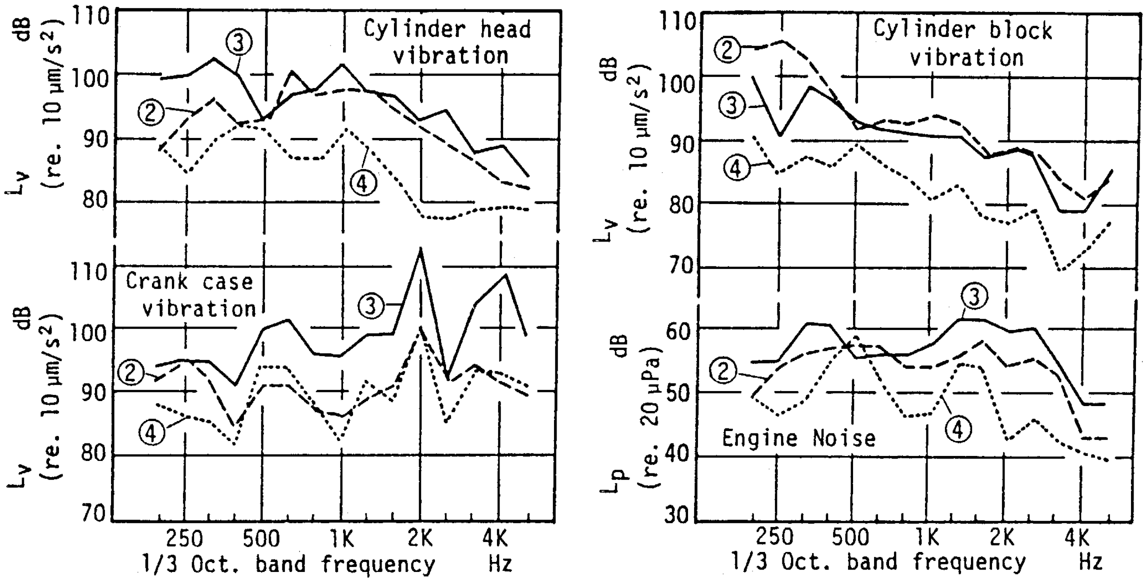


Fig. 5 Frequency characteristics of vibration and noise for each time segment (1500rpm, no load, idling).

scavenging period and ④ latter half of the scavenging period.

The vibration acceleration level of period ③ is predominantly in the frequency range of 200−400Hz at the cylinder head, in 125−200Hz at the cylinder block and in 500−5kHz at the crank case. The peak of 2kHz and 4kHz at the crank case in period ③ is particularly remarkable.

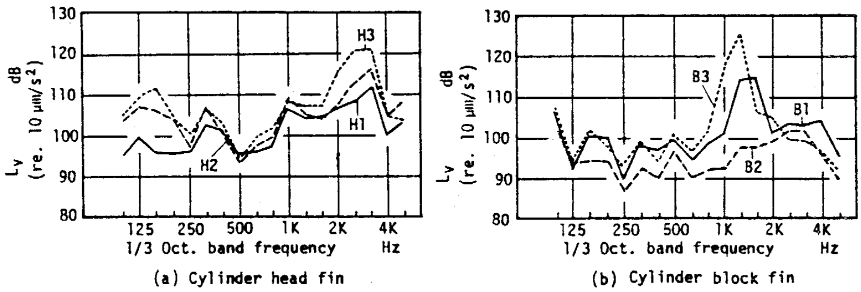


Fig. 6 Frequency characteristics of fin vibration in period ③ (head fin, block fin).

Comparing the noise radiated during each period of an engine cycle, the noise in period ③ is strongest. A good correspondence between vibration and noise is observed in wide frequency ranges, but there is still no peak in engine noise as high as the crank case vibration at 2kHz and 4kHz.

As the test engine is an air—cooled engine, there are many cooling fins outside the head and block. Both the head and block had three types of fins on them. Fig.6 shows the frequency characteristics of fin vibration in period ③.

The cylinder head fins (Type H1, H2, H3) vibrate strongly in the frequency range of 2k—3.15kHz and additionally in 160Hz, 1kHz and 5kHz. As for the block fin (Type B1, B2, B3) vibration, we can see its peak in the 1k—1.6kHz range.

From the results of a hammering test on a stationary engine, the resonance frequencies of the fins are found in the same frequency ranges as in the case of the running engine. It can be said that the main sources of idling noise of the test engine are the block fins and the cylinder head fins. They strongly vibrate in the first half of the scavenging period ③ and radiate noise in the 1.25k—2.5kHz range of their resonance frequencies.

Then it becomes possible to clarify the main noise sources and their generation times in the idling operation of a tested engine.

Vibration and Noise in Light Load Operation

Effects of Engine Load.

Fig. 7 shows the effects of load on vibration and noise of the engine in terms of the temporal average of each signal throughout the whole cycle. The vibration acceleration levels of the engine show the largest value at the cylinder head in the range of 1k—4kHz and at the crank case in the range of 1.6—5kHz, while the engine noise shows peaks in the frequency range of 2k—3.15kHz. As far as vibration level, the strongest responses are seen at the cylinder block in a frequency range below 400Hz, at the cylinder head in the middle range around 1kHz and at the crank case in a range above 2kHz. The effects of engine load on the viration and noise can be mostly seen in the range of 1.6k—4kHz.

Fig. 8 shows the noise contribution of each part of the engine for the one—third octave band frequency component of 2kHz which mostly changes its level with engine load as shown in Fig.7. This result was obtained by calculating the acoustic power

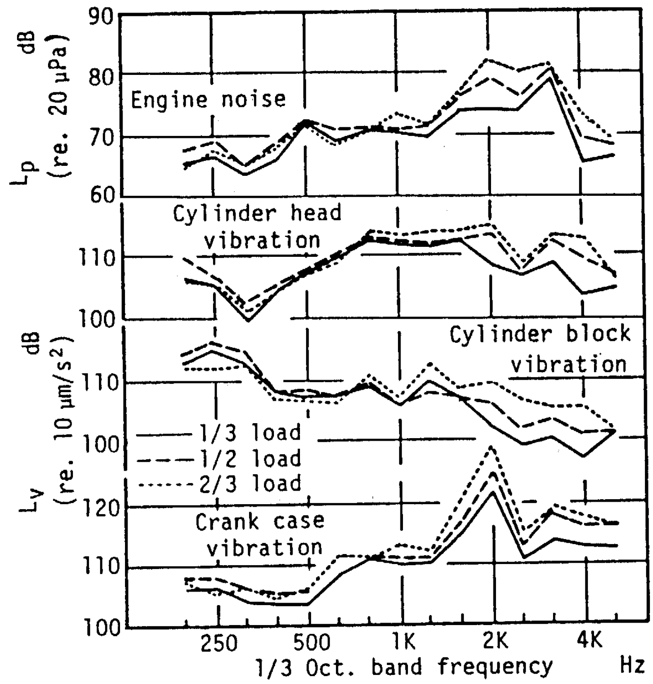


Fig. 7 Effects of engine load on Vibration and noise (2000rpm, fifth gear).

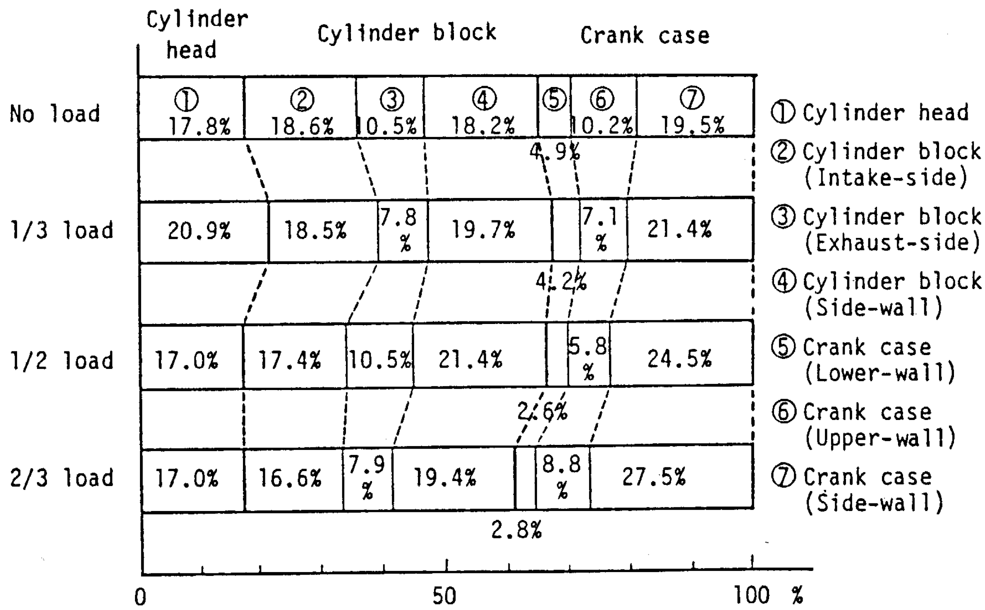


Fig. 8 Noise contribution ratio of each part of engine surface in 2kHz (fifth gear).

ratio through Sound Intensity Measurement data. The contribution ratio of the noise generated from the side wall of the crank case increases with the engine load.

Time Histories of Engine Noise and Vibration.

Fig. 9 shows the time histories of cylinder pressure, vibration accelerations on each portion and engine noise in light load operation at 2000rpm. In the vibration of the cylinder head and block, impulsive responses are observed just after Top Dead Center (T.D.C.). Judging from the wave forms of the responses at the cylinder block, this may be caused by piston slap. In crank case vibration, strong responses can be seen not only after Top Dead Center but also around Bottom Dead Center (B.D.C.).

From the results of this figure and the previous one, we can infer that the combustion impact causes crank case vibration and noise secondarily through the driving gears.

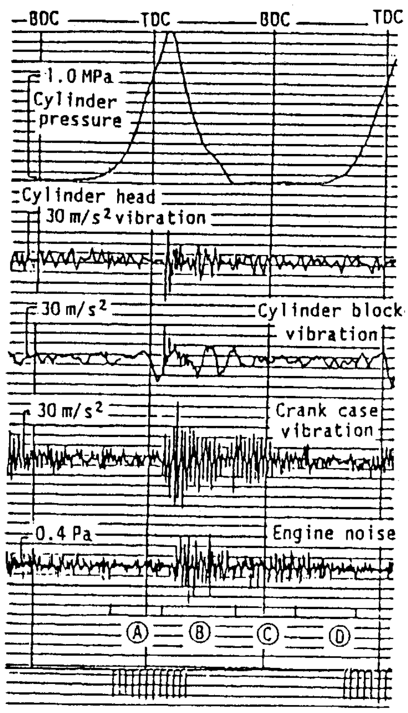


Fig. 9 Time histories of vibration and noise (2000rpm, light load).

Frequency Analysis for Each Time Segment

Fig. 10 shows the frequency characteristics of engine vibration and noise analyzed for the four segments which are shown in the bottom of Fig.9. The frequency components of noise around 2kHz are predominant and these are strongly radiated in period Ⓑ, i.e. the combustion and expansion period. From the vibration data on each portion of engine structure, the main noise source of these components seems to be the crank case.

As for the component at 3.15kHz, there is no peak in vibration outside of period Ⓑ, except in engine noise. The source of this noise component may be cylinder head fins as described in the case of idling operation.

Time Histories of Each Band Frequency Component of Noise.

The frequency analysis in terms of time average does not give information about the times when the vibrational and acoustical responses are brought about, while the time

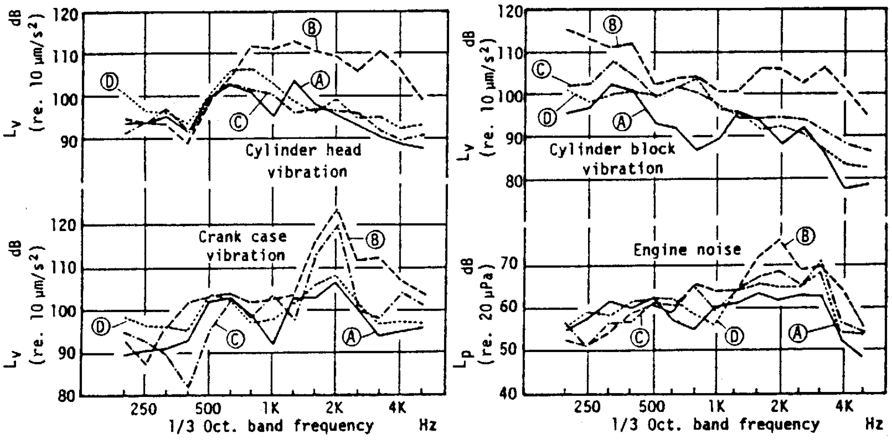


Fig. 10 Frequency characteristics of vibration and noise for each time segment (2000rpm, light load, fifth gear).

history of responses does not give the frequency characteristic. Investigating the responses through intermediate processing, such as the time history display of wide band frequency component of response signal, we can see when each band frequency component of response occurs in one cycle of a running engine.

Fig. 11 shows a typical example of the time histories of each one–third octave band frequency component of engine noise, where the ordinate expresses the intensity of noise. A strong noise radiation in the range of 1.6k–3.15kHz is seen just after T.D.C. Within this range, a component of 2kHz is predominant and the 1.6kHz and the 3.15kHz follow in this period. In the period around B.D.C., components of 3.15kHz and 2.5kHz are strongly radiated. These noises may be caused by cylinder block fins as described previously.

As found in the above example, the time history of each band frequency component of noise makes it possible to have information about the frequency characteristics radiated in a particular moment and its origin with the aid of the vibration data.

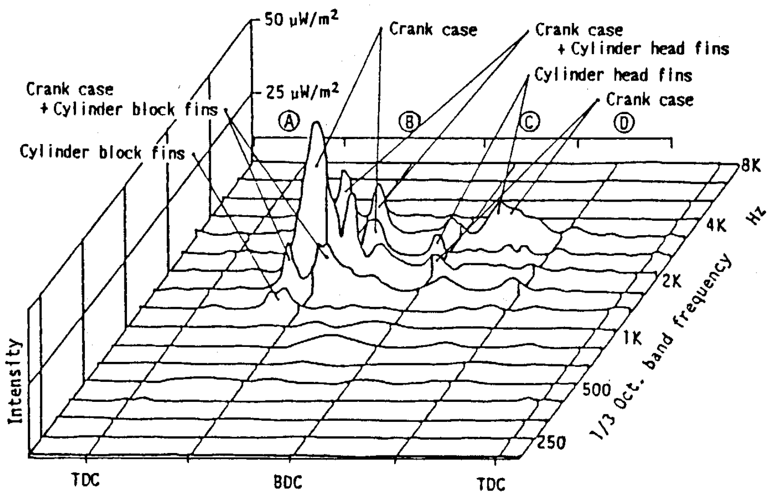


Fig. 11 Time histries of each band frequency component of engine noise (2000rpm, light load, fifth gear).

An Approach for Diagnosis of Impact Origin

An attempt was made to clarify the impulse excitation origin which causes the engine noise to be generated. In these experiments, the engine was run at 2000rpm with no load.

Vibration Isolation of Engine Parts.

The excitation origin can not necessarily be identified from the fact that a particular engine portion vibrates and/or radiates noise strongly. It is important, needless to say, to distinguish the impact origin with noise origin in proceeding with machine noise control activities.

The test engine is regarded as a construction of two large parts, i.e. the cylinder head and block and the crank case. To insulate the vibration transmitted from one to another, a rubber plate 4mm thick was inserted and bolted to a thickness of 3mm. To minimize the vibration transmission through the fitting bolts, rubber rings were also inserted at the conjunctions. Fig. 12 shows the vibration insulation effects of the rubber plate obtained by comparing the results of hammering tests with and without insertion of the rubber plate. As sufficient isolation effects are seen in the range above 1.25kHz, further discussion regarding this frequency range will follow here.

In running operation, an aluminum plate 3mm thick was also set between the cylinder block and crank case eliminating the effect of the changes of scavenging timing and compression ratio with the rubber plate insertion. The frequency characteristics of vibration accelerations and radiated noise are shown comparatively in Fig.13 with and without insertion of the plates. Large decreases in the sound pressure level of engine

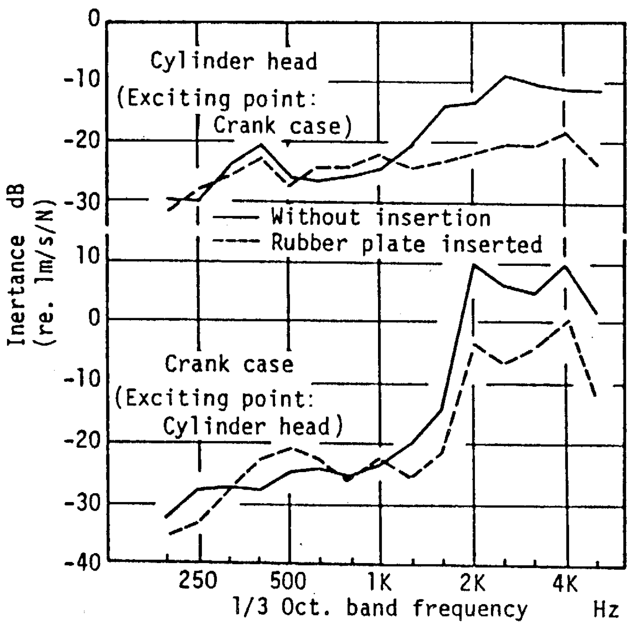


Fig. 12 Vibration insulation effects of rubber plate (Impulse hammer excitation).

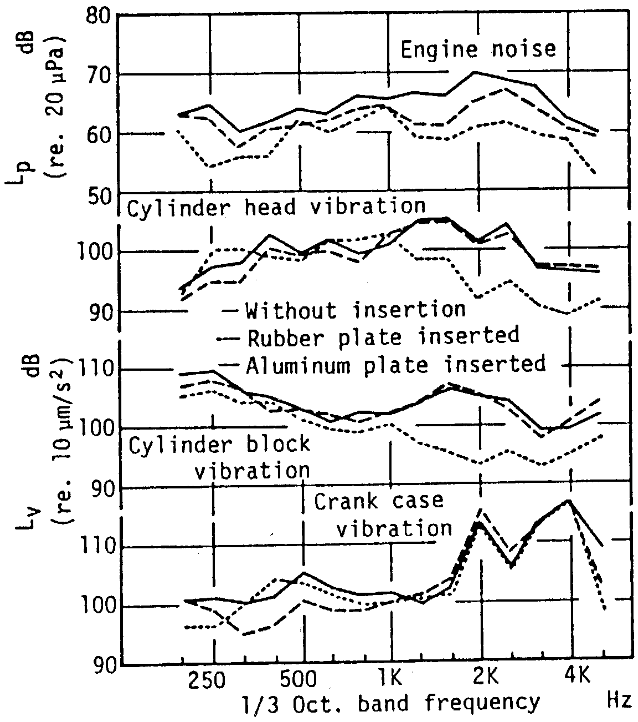


Fig. 13 Survey of impact origins through vibration isolation (2000rpm, no load).

noise and the vibration acceleration level at the cylinder head and block are seen in the range of above 1.25kHz by inserting the rubber plate, while there is no significant difference in the vibration at the crank case. These results imply that the predominant impacts are generated in the crank case, transmitted to the cylinder block and the head and radiate noise from their surfaces.

Elimination of Engine Parts.

Another easy way to identify the impact origin is to eliminate the engine parts which seem related to the generation. As described in the previous section, the main impact source of this test engine at no load 2000rpm operation is in the crank case. The most probable cause of the impulse response in the crank case is the impact of transmission gears or primary reduction gears.

Fig. 14 shows the frequency characteristics of vibration accelerations and noise in comparison with different gearings. In neutral operation, both vibration accelerations and sound pressure level are at most 3dB lower than during fifth gear (reduction gear ratio 3.08) operation. When the primary reduction gear is eliminated, these responses decrease their level still more, i.e. the sound pressure level decreases above 500Hz, the vibration acceleration levels decrease about 5dB above 1kHz at the cylinder head and block, and widens about 500Hz at the crank case. From the results of this survey of impact origin, the most predominant source of noise in this engine at no load operation seems to be the impact of the primary reduction gears.

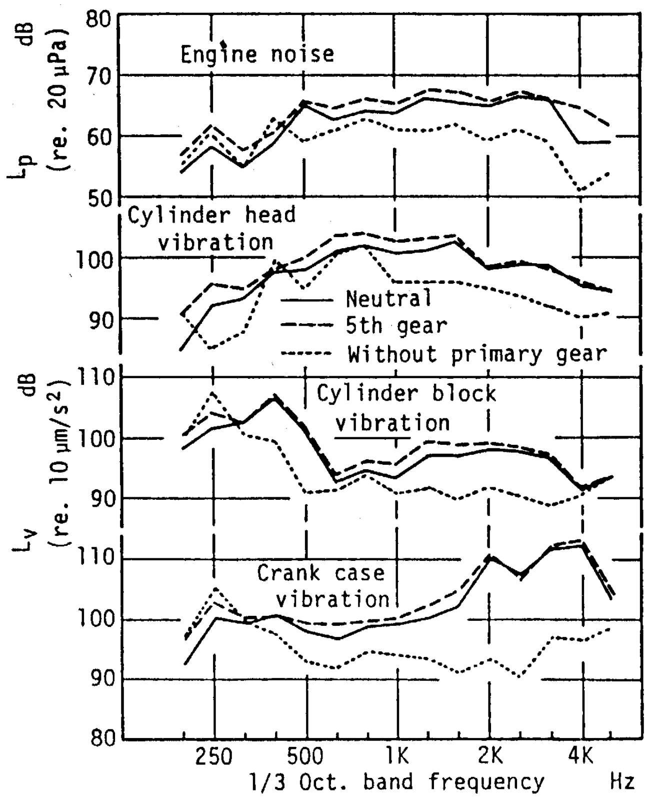


Fig. 14 Effects of gearing in drive system (2000rpm, no load).

Conclusions

(1) A method of partial frequency analysis is proposed to obtain the frequency characteristics of response signals which occur in particular time segment during one operation cycle of an engine.

(2) This technique makes it possible to know the contribution of each instantaneous response to one entire cycle of engine vibration and noise.

(3) In the idling operation of the air-cooled motor cycle engine employed here, the strongest noise radiation is seen in the first half of the scavenging period. The sources of this noise are the cylinder block in the low frequency range, the cylinder head in the middle range and crank case and cooling fins in the high frequency range.

(4) In light load operation, the noise components of 1.25k–3.15kHz are most strongly radiated just after T.D.C. from the crank case, cylinder head and cylinder block fins.

(5) As a result of a survey of impact origins in this engine, the most predominant impacts occurred in the impact of the primary reduction gears, which are transmitted through the crank case and radiate a certain amount noise from engine surface in no load operation.

(6) The proposed methods here offer promising techniques for identification of noise sources and their causes in operating engines.

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