Noise control of portable generator set*

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Major noise sources in a noisy portable I.C. engine driven generator set have been identified. The generator set engine is petrol start and kerosene run. The exhaust silencer of the engine was providing substantial insertion loss. The separation of engine combustion and mechanical noise indicated that their contribution was almost equal. Since generally combustion noise is more than mechanical noise, the main noise sources were considered to be mechanical in nature. Sound intensity measurements were performed to identify major noise sources in the generator set when the exhaust was ducted away. The results of these measurements indicated that the main sources of noise in the generator set are: cooling fan cover, silencer shell, silencer cover and the engine crankcase. Noise control measures were applied to these parts. Sound pressure and power levels were measured before and after the application of noise control measures. Constrained layer damping treatment and stiffening of the cooling fan cover had a combined effect of reducing the sound pressure level by about 3 dB(A). Rigid clamping of the silencer also reduced the noise level. A partial enclosure was designed for the generator set. The partial enclosure reduced the sound pressure and power levels by about 4 and 3.7 dB(A) respectively. There was an increase in the engine cylinder head temperature due to the enclosure, but the increase in temperature was considered to be safe. An overall noise reduction of 8.5 dB(A) was obtained on side 4 of the generator set as a result of the implementation of all the noise control measures. The noise reduction on the other sides of the generator set was also substantial.

Introduction

The first task of any noise control programme is to identify the major sources of noise radiation. The greatest reduction in noise can only be achieved by reducing the noise radiated from the most noisy sources. A reduction of noise therefore requires not only the treatment of many different sources but also calls for different approaches in each particular case depending upon the degree of the contribution of the various individual sources to the total noise and finally upon the economical aspects. The noise of vehicles and machines is mainly defined by the noise generated by their power plant which is usually an engine. One of the predominating contributors to the total engine noise is the noise emitted by the engine surfaces. Russell, 1973, Thien, 1973, and Kabele and Anderkay, 1975 have reported techniques to find the important noise radiating areas of the engine surfaces and noise reduction measures. Herbert and Russell, 1982 describe techniques for measuring combustion noise. Analysis of noise produced by piston slap and methods of its control in diesel engines has been given by Haddad and Howard, 1980.

Portable engine driven generators are used to supply electricity in shops, offices and homes when there is a break in power supply. In these generators the alternator and engine are mounted on a frame through rubber mounts as shown in Figure 1. Such generators are normally placed outside shops/offices and generate high noise levels causing annoyance to people in the neighbourhood. In the present study noise control of one such, particularly noisy generator set (Figure 1), was undertaken.

Generator noise may be controlled by reducing the excitation applied to the structure by the engine combustion process and mechanical impacts. In addition noise may be controlled at source by modifying the structure so that a) the basic engine structure accepts the forces applied to it by the combustion process and mechanical

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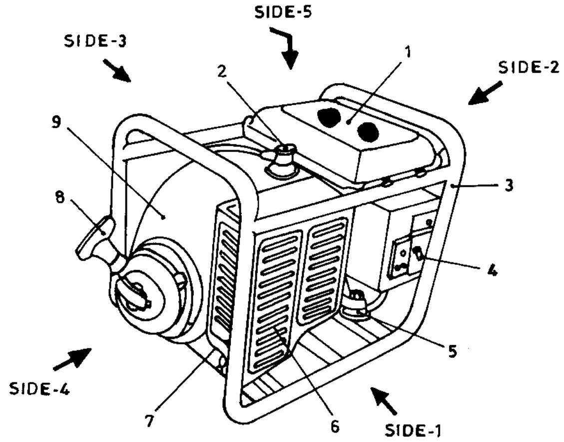

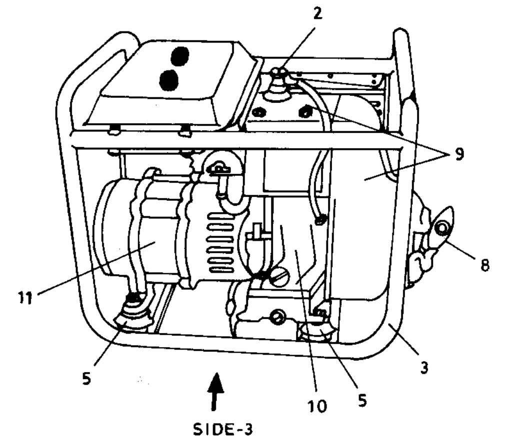

Figure 1. Portable generator set (1: Fuel tank, 2: Spark plug, 3: Frame, 4: Electric switch, 5: Rubber mounts, 6: Silencer cover, 7: Silencer, 8: Starter handle, 9: Cooling fan cover, 10: Crankcase, 11: Alternator).

impacts, with the minimum structural deflection b) the panels and covers which radiate noise are isolated from structural vibration, or designed for minimum response to vibration excitation c) the vibration energy in the normal modes of vibration is rapidly dissipated as heat by vibration damping treatments and d) designing a partial or full enclosure. The methodology for investigating, evaluating and designing controls for noise emanating from an internal combustion engine or arising from the addition of necessary component has been described by Baxa, 1982. For a reduction of 10 dB (A) or more, it is necessary to attenuate the noise of all external engine parts by a complete encasing of the engine (Thien, 1973, and Miller and Montane, 1978).

Test setup and instrumentation

Noise and vibration measurements were conducted on an I.C. engine driven portable generator with a view to identify its major noise sources for noise control. The generator had a rated AC output of 800 VA and rated frequency of 50 Hz. The engine is petrol start and kerosene run. It is 4-stroke single cylinder, air cooled having 120 cc displacement. The engine speed is 3000 rpm (3240 rpm under no load). The components of the portable generator are shown in Figure 1. The various sides of the generator are numbered as (Figure 1); Side 1: Front (control panel) side, Side 2: Side opposite to side 4, Side 3: Back (opposite to control panel) side, Side 4: Side on which starter pulley is located, and Side 5: Top (fuel tank) side.

Sound pressure levels were measured with the help of a portable precision sound level meter. For sound intensity measurements, a two-microphone sound intensity probe, dual channel FFT analyzer and desk top computer with sound intensity software were used. The sound intensity probe had ½ inch microphones separated by a 12 mm spacer. Vibration frequency response function of the structures was determined by impacting the structures with instrumented hammer which includes force transducer and line drive amplifier, and measuring the vibration response with the help of an accelerometer through a charge amplifier. Both the force and response signals were fed to the dual channel FFT analyzer to obtain the frequency response function.

Two similar generators were used in this work. The sound power of generator no. 1 was determined by measuring sound intensity and the sound power levels of generator no. 2 were measured by measuring sound pressure levels over a hypothetical hemisphere as per ISO 3745. All the measurements were performed in an acoustic chamber. The average background sound pressure level in the acoustic chamber was measured to be 25 dB(A). The load to the generator was applied with the help of electric bulbs of total 800 W. A partial enclosure was used on generator no. 2 whereas other noise control measures were applied on generator no. 1.

Measurement results on original generator

Initial noise levels of the generator were measured in order to have reference values for comparison after applying control measures. The sound pressure levels obtained at a distance of 1 m from generator no. 1 surfaces when the generator is placed in acoustic chamber with exhaust ducted out, at no load and 800 load, are shown in Table 1. The exhaust was ducted out because as shown in section 3.1, the acoustic performance of the silencer was quite good and other noise sources needed to be tackled.

Table 1. Sound pressure levels of the sides of generator no. 1 with exhaust ducted away

Load	Sound pressure level, dB(A)			
	Side 1	Side 2	Side 3	Side 4
No load	86.0	84.5	85.0	85.0
800 W	88.5	86.5	85.0	86.5

Sound power levels of this generator were measured to be 91.2 and 92.0 dB(A) at no load and 800 W load respectively.

The sound pressure levels of the generator set without exhaust ducted away were also measured and are shown in Table 2.

Table 2. Sound pressure levels of the sides of generator no. 1

Load	Sound pressure level, dB(A)				
	Side 1	Side 2	Side 3	Side 4	Side 5
No load	86.5	85.5	85.0	87.0	86.5
800 W	89.5	87.5	87.5	88.5	87.0

Exhaust silencer insertion loss measurement

Insertion loss, the difference between the sound pressure levels at an external reference point with and without silencer in place, can be evaluated experimentally from simple sound pressure measurements. Insertion loss measures the effectiveness of the silencer as it operates in a particular installation, including the influence of coupling to the engine source and the tail pipe. Insertion loss of the silencer was measured at no load and at 800 W load conditions by measuring A weighted sound pressure levels at 1 m distance from the end of a long pipe connected to the exhaust of the engine with and without silencer. In no load and in 800 W load conditions the insertion loss measured was 27 and 28 dB respectively as shown in Table 3. This is substantial noise attenuation indicating good performance of the silencer.

Table 3. Insertion loss measurement

Load	Sound pressure level Without Silencer (X)	With silencer (Y)	Insertion loss X-Y
No load	95	68	27
800 W	97	69	28

Separation of combustion and mechanical noise

In a running internal combustion engine, excitations responsible for the generation of engine noise are quite complex. During an engine cycle several impulsive inputs act at various points of the engine structure arising at different instants in rapid succession. These energy inputs travel along several structural paths getting attenuated in the process, eventually reaching the exterior surface of the engine where it is radiated as noise to the surroundings. There are two primary sources of engine noise – combustion and mechanical. Prior to deciding the noise control strategies to be employed, the contribution of combustion and mechanical noise are to be investigated. Combustion and mechanical components of engine noise can be separated by ‘single explosion test’, ‘motored engine test’ or ‘running engine test’ (swinging the injection timing). Motored engine test is a simple one and does not need major changes in the engine and sophisticated instruments. This method was used in the present work to assess the contribution of mechanical and combustion noise.

The generator set was connected to a 1 hp motor mounted on a bed. The starting unit of the generator set was removed and a pulley was fixed on the shaft of the engine. The total sound power level measured due to combustion and mechanical noise was 92.8(A) and the sound power level due to mechanical noise alone (motored test) was measured to be 89.7 dB(A). So the percentage power contributed by the combustion and mechanical components is 50.9% and 49.1% respectively. Usually in an internal combustion engine, the contribution of combustion component is much more as compared to the mechanical component. But in this case it is found that mechanical and combustion components share an almost equal amount of noise. So it can be concluded that the mechanical noise is more than usual and that there is a need to control mechanical noise.

Noise Ranking of Generator Sides

Sound intensity measurements were performed to identify the generator sides producing high noise. Scanning method was used to obtain the sound intensity of five sides of the generator at 800 W load. During the measurements, the exhaust was ducted out of the acoustic chamber with the help of a long pipe, so that the exhaust noise does not interfere with the noise generated by other parts of the engine. The microphone probe was slowly moved over a side of the generator. The information regarding the area of the sides and frequency range was fed to the computer and the sound power rankings of the sides obtained is shown in Figure 2.

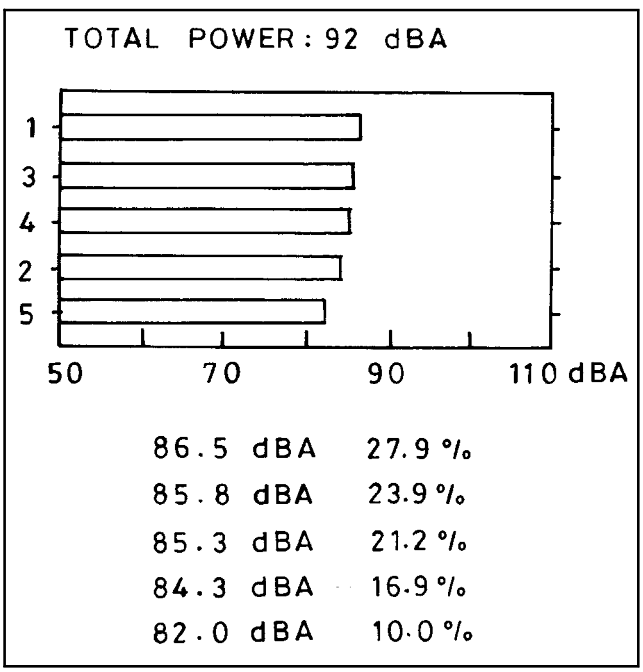

Figure 2. Sound power ranking of generator sides

Results show that sides 1,3 and 4 produce high noise. The highest noise is generated by the front side (side 1) of the generator. It was decided to obtain the sound intensity maps of the three sides generating high noise.

Sound intensity mapping and spectra of generator

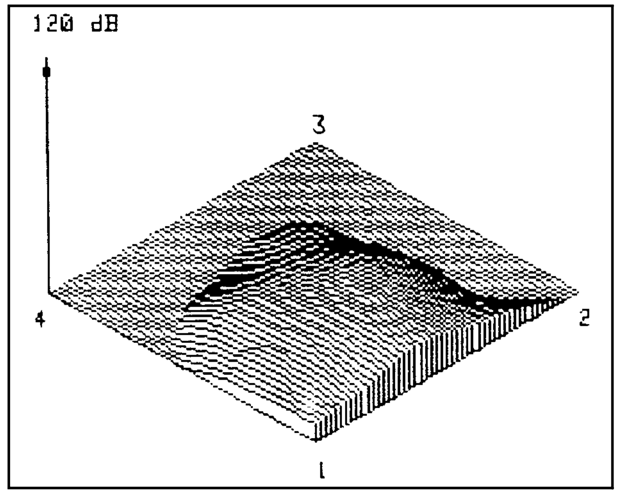

Figure 3. Sound intensity map of the front side (side 1) of the generator

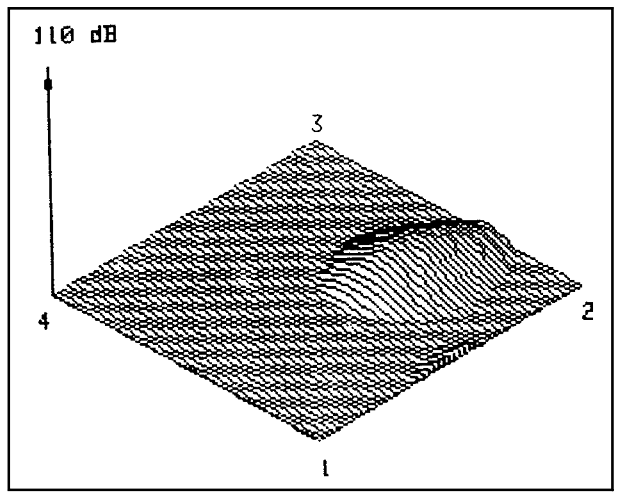

Figure 4. Sound intensity map of the back side (side 3) of the generator

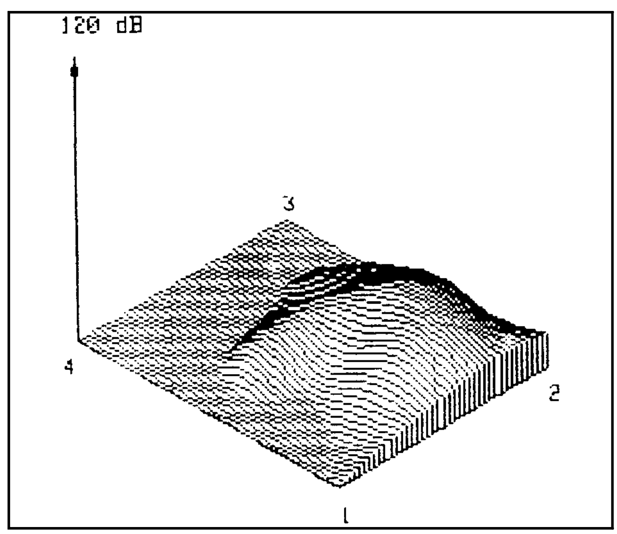

Figure 5. Sound intensity map of the side 4 of the generator

Sound intensity maps of the three sides of the generator were obtained. A grid of 6 rows and 6 columns (i.e. 36 points) was made on side 1 and side 3 whereas on side 4, a grid of 6 rows and 5 columns (i.e. 30 points) was made. Sound intensity was measured at each of these points of the grids at 800 W load. The information regarding the area per point, frequency range, number of rows and columns, number of averages etc. were fed to the computer. The 3-dimensional maps of A-weighted sound intensity are shown in Figures 3 to 5. The base level in these maps is 95 dB(A) and the lines 1-2 represent the bottom and lines 3-4 the top of each side.

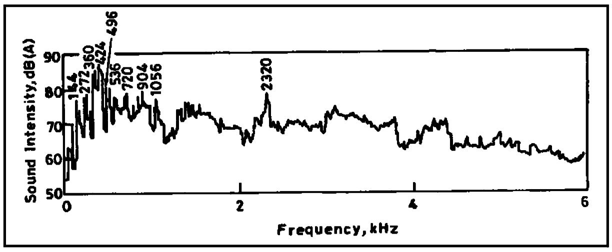

Figure 6. Frequency spectra of sound intensity on side 1

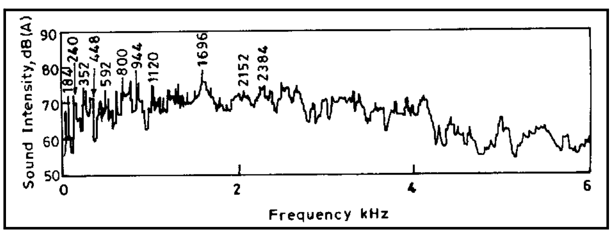

Figure 7. Frequency of sound intensity on side 3

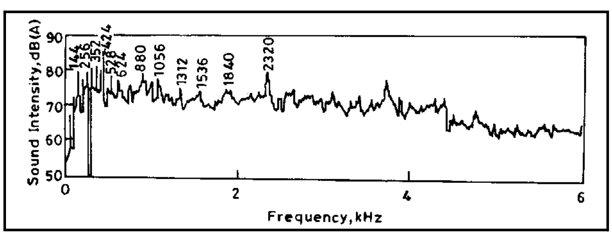

Figure 8. Frequency spectra of sound intensity on side 4

The results of side 1 indicate that the intensity level is high in the region of engine and silencer cover and silencer (Figure 3). Figure 4 shows that on side 3, high noise is generated in the region of engine crankcase and cooling fan cover. On side 4 the maximum level is in the region of cooling fan cover and silencer (Figure 5). Figures 3 & 4 do not show high noise in the region of engine cylinder head which would normally be encountered if combustion noise were dominant. In the engine, the crankcase region is generating high noise (Figures 3 & 4).

Frequency spectra of sound intensity were obtained on the sides of the generator set at points where high noise levels were observed in earlier measurements. The sound intensity probe was held normal to the point at which frequency spectra was desired and sound intensity was averaged over 25 samples. A-weighted sound intensity frequency spectra obtained are shown in Figures 6 to 8. These spectra were obtained when the generator set was running at 800 W load.

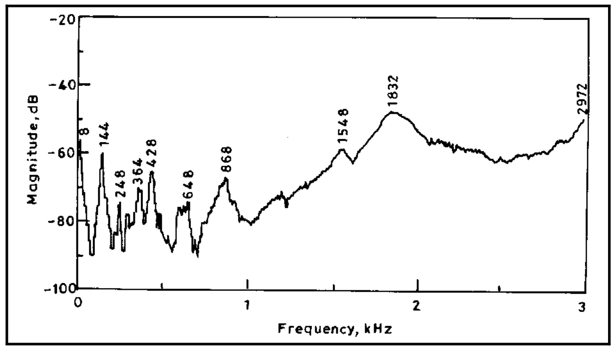

Figure 9. Frequency response function of silencer

Frequency response function of structural components

Any finite structure has an infinite number of modes of vibration. Some of these may be in a frequency range in which significant excitation occurs. Thus they may magnify the forced response of the structure and generate high levels. The modes depend on the mass and stiffness of the structure and the forced response depends on the damping as well. Frequency Response Function is basically a description of the output of a system in terms of the input of the system. In structural dynamics work this usually takes the form of the vibration response (in terms of displacement, velocity or acceleration) at some location on the structure relative to the force applied at same or another point on the structure. The frequency response function for mobility of silencer, silencer cover, cooling fan cover and crankcase were obtained when the engine was not running. Hence an indication of the influence of the dynamic properties on the radiated noise could be found by comparing spectra of acoustic intensity and mobility. The result of experiments are shown in Figures 9 to 12 and the spectra have been compared in the next section.

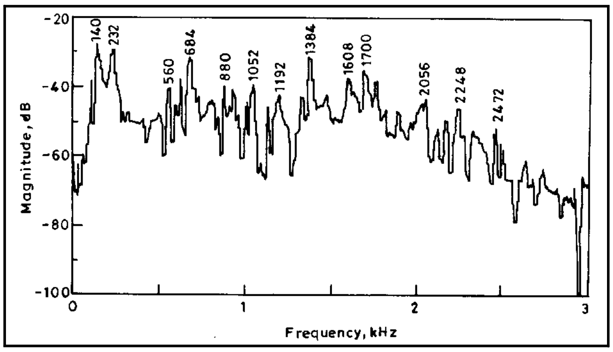

Figure 10. Frequency response function of silencer cover

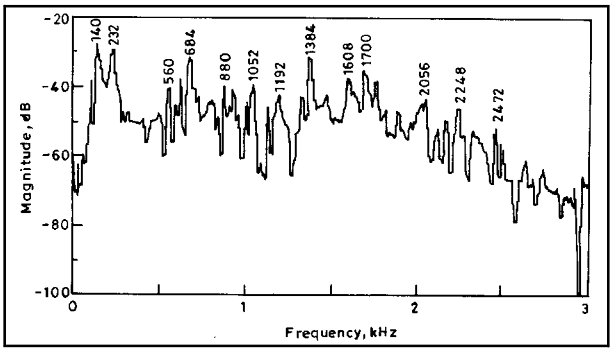

Figure 11. Frequency response function of cooling fan cover

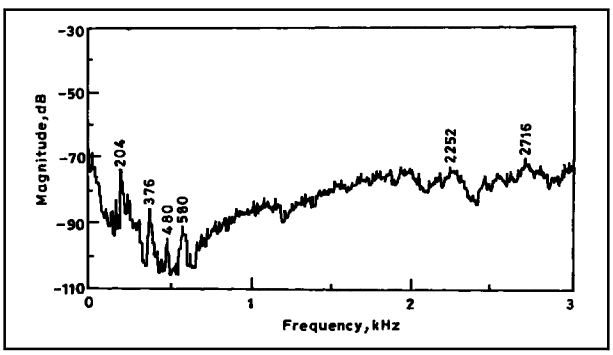

Figure 12. Frequency response function of crankcase

Noise Source Identification

The excitation frequencies which can excite the system and as a result give rise to high peaks in sound intensity spectra have been identified. One of them is the firing frequency which depends upon the engine speed. Others are due to fans and gear mesh. The excitation frequencies due to fan is governed by the number of blades and the engine speed since every time a blade passes a given point, the air at that point receives an impulse. Frequencies due to gear mesh depend upon the number of teeth and shaft speed. The excitation frequencies associated with the running of the generator are given in Table 4.

Table 4. Excitation Frequencies

Source	Excitation frequency
Firing frequency 25 Hz (at full load) and harmonics	27 Hz (at no load)
Blower/cooling fan frequency	800 Hz (50 x 16)
Gear Mesh frequency	1050 Hz (50 x 42)
Fan blade pass frequency	1000 Hz (50 x 20)

A comparison of sound intensity spectra at different points on three sides of the generator with the Frequency Response Function of various structural components and with excitation frequencies is carried out in order to find the sources radiating high noise.

On side 1 main peaks in the sound intensity spectra occur at 144, 272, 360, 424, 496, 536, 720, 904, 1056 and 2320 Hz (Figure 6). When compared with structural resonance frequencies of crankcase, cooling fan cover, silencer and silencer cover (Figures 9 to 12) and with the excitation frequencies (Table 4), it was found that the peaks at 144, 360 and 424 Hz could be because of the silencer and the peaks at 536 and 1056 Hz could be because of silencer cover and cooling fan cover.

Figure 7 shows that on side 3 some of the main peaks occur at 184, 240, 352, 448, 592, 800, 944, 1696 and 2152 Hz. Some of these peaks are very close to the peaks observed in the frequency response function of cooling fan cover and crankcase and could be due to resonances of these components. The peak at 800 Hz appears to be due to the cooling fan itself.

The main peaks in sound intensity spectra of side 4 are observed at 144, 256, 352, 424, 528, 624, 880, 1056, 1312, 1536, 1840 and 2320 Hz (Figure 8). On comparison of these peaks with Figures 9 to 11, it can be said that the peaks at 144, 256, 424, 624, 1536 and 1840 Hz are most likely because of silencer, whereas the peaks at 528 and 1056 Hz could be because of cooling fan cover. The silencer cover could be responsible for peaks at 880 and 1312 Hz.

The preceding discussion indicates that the cooling fan cover, silencer shell, silencer cover and the engine crankcase of the generator are radiating high noise.

Noise Control Measures on Generator Components

Many factors influence the selection of noise control methods. These may be of either acoustical, mechanical, practical or economical character. The noise control measures applied to the cooling fan cover and silencer shell are described in the following sections.

Cooling Fan Cover

Damping Treatment

Damping treatments are often used to solve resonant noise and vibration problems, especially those associated with sheet metal structures. These involve use of high damping polymeric materials bonded to the sheet metal structures in unconstrained or constrained layer types of treatments. In the unconstrained type, the polymeric viscoelastic layer undergoes alternate tension and compression with the bending vibration of the sheet metal dissipating vibratory energy. While in the constrained type, the polymeric viscoelastic layer is constrained between the sheet metal and a metallic constraining layer, inducing shear in the viscoelastic layer and thus resulting in dissipation of vibratory energy. The latter configuration is generally seen to be more effective in providing damping for a given increase in size or mass of the system. It is also seen that a symmetrical configuration with constraining layer with the same thickness and material as of the sheet metal structure and use of a high value of thickness of the viscoelastic layer with high material loss factor, provides maximum possible damping. Earlier measurements indicated that the cooling fan cover was resonating. To suppress the resonating vibrations of cooling fan cover, constrained-layer damping treatment was applied to it. Keeping the above mentioned factors in view, a PVC sheet of 3 mm thickness was used as damping layer with a mild-steel sheet of 1.15 mm as a constraining-layer. The damping loss factor of PVC used was 0.5 to 0.6 at room temperature in the frequency range of 20 to 300 Hz, and its shear modulus varied from 1.4×10^7 N/m^2 to 2×10^7 N/m^2 in the above range. The adhesive used was “Araldite”.

Initially the constrained damping treatment was applied on top and left side of the cooling fan cover. The results were encouraging, as sound pressure level decreased by 1 dB(A). The constrained-layer damping treatment was then, applied on the remaining parts of the cooling fan cover also. Table 5 shows the sound pressure levels obtained with this treatment. It is seen that the levels have decreased by 1.5 to 2.0 dB(A) on different sides of the generator set. The sound power level decreased from 92.0 to 90.7 dB(A) (Table 6).

Table 5. The effect of noise control measures on sound pressure levels of generator no. 1 at 800 W load with exhaust ducted away

S.No.	Noise control measures	Sound pressure level, dB(A) Side 1	Side 2	Side 3	Side 4
1.	None	88.5	86.5	85.0	86.5
2.	Constrained layer Damping on fan cover	86.5	85.0	83.0	85.0
3.	Constrained layer damping on fan cover and silencer held rigidly with external support	85.5	84.0	84.0	84.0
4.	Fan cover damped and Stiffened	84.5	84.0	83.0	84.0

Stiffening of Damped Cooling Fan Cover

Stiffening is effective in increasing the natural frequencies of the component. Thus by stiffening, the resonances can be avoided and hence the noise levels can be brought down. Increasing the natural frequencies can also be an advantage if it then falls within a range where the exciting forces are less. Thick plates of mild steel were joined with “m-seal” on the inner side of the cooling fan cover. The effect of damping and stiffening of cooling fan cover on sound pressure and power levels is shown in Table 5 and 6 respectively. The results at S.No. 4 are to be compared with the results at S.No. 2 in these Tables. The results indicate a further decrease of about 1 dB(A) in sound pressure levels and 0.5 dB(A) in sound power levels.

Table 6. The effect of noise control measures on sound power level of generator no. 1 at 800 W load with exhaust ducted away.

S. No.	Noise control measurea	Sound power level, dB(A)
1.	None	92.0
2.	Constrained layer damping on fan cover	90.7
3.	Constrained layer damping on fan cover and Silencer held rigidly with external support	90.3
4.	Fan cover is damped and stiffened	90.2

Silencer

Noise radiation from the silencer shell and related piping or ‘shell noise’ is a problem that has recently grown in importance as outlet noise levels have been reduced. In the generator set, it was found that silencer was one of the major noise radiating sources apart from cooling fan cover and crankcase. The noise radiated by the silencer can be reduced by controlling its vibration by modification of the dynamic characteristics by changing stiffness or mass of the structure, by isolation of the source of vibration from the body of the noise radiating structure, or by use of damping materials. In the case of silencer there is limitation on applying damping material because of high temperature.

The silencer can not be clamped rigidly with the generator set frame as this will make the rubber mounts ineffective. In order to establish that clamping of silencer gives rise to reduction in noise level, it was clamped rigidly with an external support temporarily. This resulted in reduction of around 1 dB(A) in sound pressure level on sides 1, 2 and 4 (Table 5). The sound power level of the generator set reduced from 90.7 to 90.3 dB(A) as shown in Table 6.

Partial Enclosure Design and Performance

Noise can be controlled by modifying the acoustic transmission path between the source of the noise and receiver. Enclosure is one of the effective means to modify the transmission path of sound. In the case of an engine driven generator set, cooling of the engine is important. So partial enclosure can be used to reduce noise level. Such an enclosure was designed and used on generator no. 2. While designing an enclosure, two types of enclosure resonances should be considered. The first is the structural resonance of the enclosure panels while the second is acoustic resonance of the air space between an enclosed engine and the enclosure walls.

Standing Wave Frequencies

Inside the enclosure a reverberant sound field is produced in addition to the noise from the source. Also the acoustic resonances occur at the standing wave frequencies which can be calculated. The standing wave frequencies for different distances between source and panel at 303°K were calculated as 8725, 4362.5, 2908.3, 2181.2 and 1745 Hz for distances of 2, 4, 6, 8 and 10 cm respectively. To suppress the acoustic resonances, absorbing material can be used on the inside of the enclosure. Absorbing material has three effects (a) it reduces the amplitude of the standing waves, (b) it raises the frequency of all standing waves resonances and (c) it broadens the standing waves. The layer of sound absorbing material should be about half the thickness of the air space to damp out the resonance considerably. In the proposed enclosure the gap between engine surface and enclosure varied from 2 to 8 cm at various points. A uniform layer of foam of thickness 2.5 cm was applied on the inside of the enclosure. The absorption coefficient of foam material varies from 0.08 to 0.75 in the frequency range of 125-4000 Hz.

The sound pressure spectra of the sides of the generator set showed that high noise mainly in the frequency range 265 to 555 Hz was present. Noise spectrum of side 4 is shown in Figure 13. The standing wave frequencies calculated in Table 7 are far above the frequencies of generator noise.

Panel Resonance Frequencies

The enclosure should be designed so that the resonance frequencies of its constituent panels are not in the frequency range in which appreciable sound attenuation is required. If the sound source radiates predominantly high frequency noise, then an enclosure with low frequency panels is recommended, implying a massive enclosure. On the other hand if the sound radiation is predominantly low frequency in nature then an enclosure with high resonance frequencies is desirable, implying a stiff but not massive enclosure.

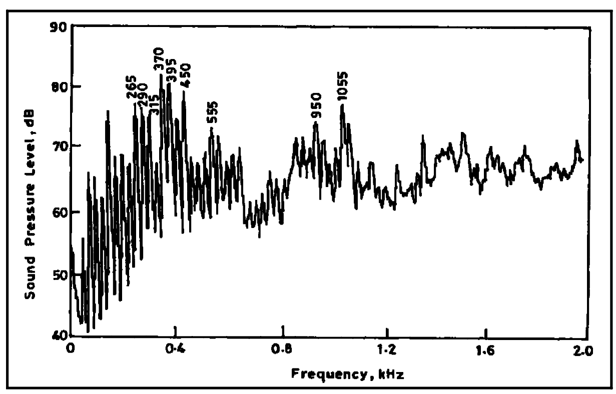

Figure 13. Sound pressure spectrum of side 4 of the generator at 800 W load.

The sound pressure spectra of generator (Figure 13) showed that noise, mainly in 265 to 555 Hz range was predominant. A light weight stiff enclosure with sandwich type walls was designed. The enclosure had to be light weight, since the generator is portable. A panel of two 20 gauge thick mild steel sheets with 5 mm thick thermocole sandwiched in between was used. The size of the enclosure was decided as per the dimensions of the generator set frame. Figure 14 shows a photograph of the enclosure on the generator set. The enclosure panels and the generator frame were isolated using rubber pads. The enclosure covered the engine, silencer and cooling fan cover portions (i.e. about two-third of the generator set). The fuel tank, alternator and front control panel portions were not covered by it.

Enclosure Performance

The performance of the enclosure was evaluated by measuring the sound pressure and power levels on the sides of the generator set with and without enclosure. The temperature inside the enclosure on the engine cylinder head was measured. The reduction in sound pressure levels obtained because of the enclosure are given in Table 7. It is found that the sound pressure level is reduced by 3.5 and 5.5 dB(A) on sides 1 and 4 respectively, at no load. The corresponding reduction at 800 W load is 3.0 and 4.0 dB(A). There is no reduction in sound pressure level on side 2 because the enclosure does not cover that side. The total sound power levels of the generator reduced by 3.4 and 3.7 dB(A) at no load and at 800 W load respectively.

Table 7. Reduction in sound pressure level of generator no. 2 due to enclosure

Load	Reduction in sound pressure level, dB(A)			
	Side 1	Side 2	Side 3	Side 4
No load	3.5	0.0	2.0	5.5
800 W	3.0	0.0	1.5	4.0

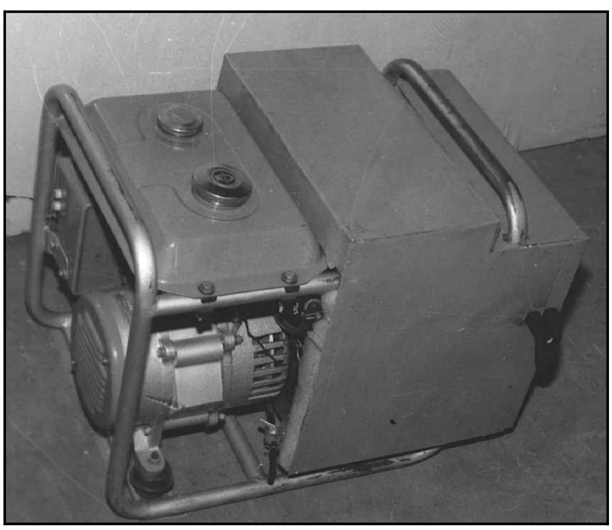

Figure 14. Partial enclosure on the generator set

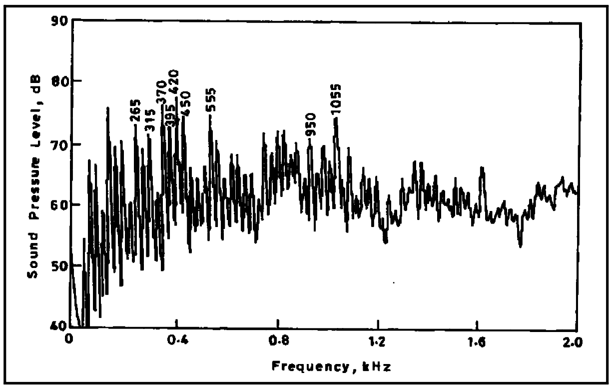

Figure 15. Sound pressure spectrum of side 4 of the generator at 800 W load.

The sound pressure spectrum of side 4 of the generator set with enclosure is shown in Figure 15 for comparison with Figure 13 which shows the spectrum without enclosure. It is seen that there is a marked reduction in noise levels at frequency range 265 to 555 Hz which was predominant in Figure 13. Noise levels at higher frequencies have also reduced.

The temperature was monitored on the cylinder head and are given in Table 8.

Table 8. Temperature on the cylinder head

Time (min)	Temperature (°C)			
	With enclosure		Without enclosure	
	No load	800 W load	No load	800 W load
0	2.31	15.1	23.3	23.2
5	68.5	73.8	54.8	61.8
10	79.8	91.4	64.4	73.3
15	87.1	102.6	66.5	77.0
20	85.8	109.3	67.5	78.1
25	87.9	111.7	68.2	78.7
30	90.6	111.5	68.5	79.1
T_{max1}	91.8	111.7	68.5	79.3
T_{max2}	122.5	145.7	93.1	103.6

T_{max1} = Maximum temperature in running condition
T_{max2} = Maximum temperature after stopping engine.

It is seen that due to the presence of the enclosure, the temperature is increased by 23.3°C & 32.4°C in running condition at no load and at 800 W load respectively and reaches 122.5°C & 145.7°C after engine has stopped at no load and at 800 W load conditions. The maximum allowable temperature of cylinder head of an aircooled engine is about 300°C (Maleey, 1995).

The maximum allowable temperature of the cylinder wall is about 127°C as given by

Kays, 1989. Because of the enclosure, the cylinder head temperature reaches maximum 112°C in running condition which can be considered to be safe.

Overall Noise Reduction

The partial enclosure was also used on generator no. 1 and the sound pressure levels of its sides were measured without ducting out the exhaust noise. These measurements were performed after other noise control measures (on the cooling fan cover) on generator no. 1 were implemented. These results of these measurements are shown in Table 9. When these results are compared with the results of Table 2, it is seen that an overall noise reduction of 8.5 and 8 dB(A) is obtained at 800 W load on sides 4 and 5 respectively, with all the noise control measures implemented on the generator set. The noise reduction on side 2 is very small because the partial enclosure does not cover that side at all. The use of a suitable unconstrained damping treatment on the crankcase, may help in further reduction to a certain extent.

Table 9. Sound pressure levels of the sides of generator no. 1 with damped and stiffened cooling fan cover and with enclosure.

Load	Sound pressure level, dB(A)				
	Side 1	Side 2	Side 3	Side 4	Side 5
No load	83.0	85.0	81.0	79.0	79.0
800 W	84.5	85.5	81.0	80.0	79.0

Conclusions

The acoustic performance of the exhaust silencer of the generator is quite good as it gives a high insertion loss. The motored engine test and the sound intensity maps of the front and the back side of the generator show that the engine noise is mainly mechanical in nature. The front, back and the pulley (fan) side of the generator are producing high noise levels. The sound intensity maps of the generator sides and sound intensity spectra indicate that the main sources of generator noise are the cooling fan cover, silencer shell, silencer cover and the engine crankcase.

The application of constrained layer damping treatment on the cooling fan cover reduced the sound pressure level by up to 2 dB(A). Stiffening of the cooling fan cover was also effective in bringing down the noise level further by 1 dB(A). Rigid clamping of silencer reduces the noise level by 1 dB(A).

The partial enclosure on the generator set reduced the sound pressure levels by up to 4.0 dB(A) in loaded condition. The reduction in the sound power level due to the enclosure was 3.7 dB(A) at 800 W load. The maximum temperature of the engine cylinder head due to enclosure was measured to be 112°C which is lower than the permissible limit.

The overall noise reduction as a result of all the noise control measures was up to 8.5 dB(A) on side 4 of the generator set.

Acknowlegement

The authors are grateful to the Council of Scientific and Industrial Research (CSIR), India for funding this research work. The contribution of D.R. Ubhe and N.K. Killa is acknowledged. This is a modified version of a paper which appeared in Applied Acoustics, vol. 55, no. 4 by N. Tandon, B.C. Nakra, D.R. Ubhe and N.K. Killa entitled “Noise control of engine driven portable generator set”, pp 307-328, 1998 and is presented with permission from Elsevier Science.

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