Design and optimization of a new range of low sound emission generating set

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This work provides an integrated approach developed with the aim of reducing the noise of generating sets, and the results obtained. The methodology and the procedures employed, with the indications obtained from theoretical analysis, are extendable to a vast range of industrial machines. The approach described has been applied to some PRAMAC Lifter generating set, presenting a high value of sound power, in order to reduce it accordingly with the value provided by the recent normative. Two production series generating sets were examined. The first, named S 12000, is a petrol engine generating set, with a 10,8 KVA alternator power and 614 cm^3 of cubic capacity; the second, named S 6000, is a diesel engine, with a 5,5 KVA alternator power and 406 cm^3 of cubic capacity. The implementation of the method described in the paper concerns the design of a new frame for the two machines and the development of new acoustical muffler.

INTRODUCTION

An engine is a typical noise source radiating multiple tones at the fundamental firing frequency. Furthermore exhaust noise of an internal combustion engine is usually composed of the most dominant frequency components directly related to engine revolution and wideband noise. For instance, the four-cylinder gasoline engine has dominant components, which are double and quadruple of the *rpm* frequency, and mid-to-high frequency wideband noise. Acoustical enclosures constitute the most frequently used measures to reduce the noise radiated by engines [1, 2]. The construction of a noise abatement enclosure is usually seen by engineers and managers, as the easiest and most cost effective method of reducing machinery noise [3]. The greatest reduction in noise can be achieved by reducing the noise radiated from the most noisy sources. Therefore a reduction of noise requires the call for different approaches depending upon the degree of the contribution of the various individual sources to the total noise [4]. For reducing exhaust noise, muffler design is also an appropriate method leading relevant noise reduction. Some well known techniques for detecting and measuring the noise radiating area of generating sets have been reported in literature [5–9].

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1. OBJECTIVES

The main purpose of this work is to design and implement an integrated approach to control and reduce the noise radiated by two differently sized and motorized generating sets. Rather than discussing theoretical aspects, this paper focuses on the technical aspects of the design presenting the technical changes developed on the generating sets in order to reduce the noise below the decibel limit stipulated in the directive 2000/14/CE.

The work was developed within a research programme entitled “Development of a new range of low noise level generating sets”, following a convention between the Industrial and Mechanical Department, University of Florence (Italy) and the company PRAMAC Lifter S.p.a., one of the leading companies in this sector. The company wants to reduce the noise of the machines almost 2 dB below the decibel limit stated before, in order to satisfy its customer request.

Two similar production series generating sets were examined: the PRAMAC Lifter S 6000 and the S 12000 generating set (Figure 1).

Figure 1. PRAMAC Lifter S 6000 and S 12000 Generating Set

The first is a petrol engine generating set, with a 10,8 KVA alternator power and 614 cm^3 of cubic capacity. The second is a diesel engine, with a 5,5 KVA alternator power and 406 cm^3 of cubic capacity. Both the generating sets, to which the methodology is applied, are open type machines and therefore not silent. The weight bearing structure is obtained by means of a tubular frame; the engine-alternator group is positioned on across a silent block. The plated hood covers the alternator and protects the electric control panel.

The two machines are particularly noisy machines, as reported in Table 1. In Table 1 is also given evidence to the obtained results (as discussed later) in terms of sound power reduction (in dB(A)).

Table 1. Generating set sound power [dB(A)]

	Diesel S 6000	Petrol S 12000
Initial conditions	104	102
Directive 2000/14/CE year 2002	98	99
Directive 2000/14/CE year 2006	96	97
Obtained results	**94**	**93**

2. EXPERIMENTAL MEASUREMENT

Particular attention has been afforded, from the first phase of the research, to the use of suitable methodologies for the study of the vibro-acoustic behavior of generating sets.

This activity can be summarized in the following points:

1. Intensimetric measurement to detect the noise sources and the sound power emission.
2. Sound power mapping to localize the sound sources.
3. Sound pressure measurement.
4. Vibration measurement to evaluate the radiated sound power from the panels and for the silent block design.

2.1. Intensimetric measurement and sound power mapping

The initial study of the generating sets, carried out by intensimetric measurements, was necessary for identifying the main sources of noise radiated by the machine and their contribution to the total noise level in order to evaluate the interventions to be undertaken [10]. In particular the experimental measurements are performed accordingly to the ISO 9614-1/98 [11] normative. According to the standard normative ISO 8528-10/98 [12] the acoustic measurement was performed when the engine reached 75% of power in a stationary condition.

All the measurement were performed in an open space (a large square near the PRAMAC laboratory). The environmental temperature was about 18 – 20°C while the environmental pressure was about 987 kPa.

For sound intensity measurement, a two-microphone sound intensity probe, B&K 4128, dual channel FFT analyser and a PC were used. Pistonphone (B&K 3541) was used for calibration. The sound intensity probe had the microphones separated by a 50 mm spacer for low frequencies and a 8.5 mm spacer for high frequency analysis. Choosing a finite number of discrete points of measurement, the time averaged rate of flow of sound energy through an element of a measurement surface is evaluated and a spatial colour map of the sound power is made. Superimposing the colour map on the machine it is possible to identify the sources of the most relevant noise. In Figure 2, the sound intensity colour map, obtained by interpolating the sound intensity values acquired at the point of measurement, of the generating sets S 6000 is shown as an example. The high intensity sound zones for the machine S 6000 are localized near the motor starter, near the motor casing and near the muffler. For machine S 12000 the highest sound levels are found near the motor casing and the muffler.

Such sources, if necessary, can be analyzed in even greater detail by increasing locally the number of discrete points or, when possible, examining the emissions of one component at a time. The sound pressure measurements have been frequently used to verify the benefits produced by the interventions undertaken.

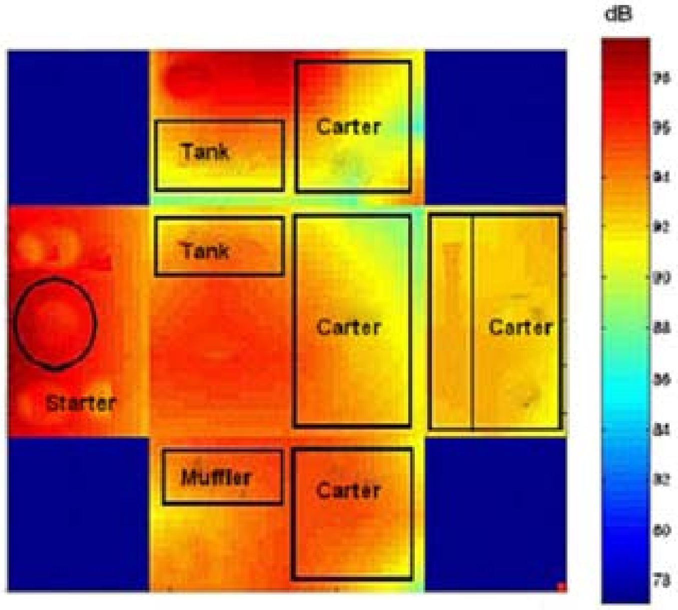


Figure 2.
Sound intensity map for the
S 6000 machine

2.2. Sound pressure and vibration measurement

The sound pressure measurement, according to the normative, standard ISO 3744:1994 [13], are not useful in the preliminary phase of the work. This is due to the fact that these measurements are not detailed enough for design purposes especially when the hypothetical sound sources are particularly close to one another. Nevertheless, these measurements have proved useful in simplifying and reducing the verification times in the subsequent phases of the research, in verifying the developed solutions and in validating the results in accordance with the Directive 2000/14/CE. In Figure 3 the sound pressure level for S 6000 and for S 12000 without (magenta) and with (blue) the designed carter is shown.

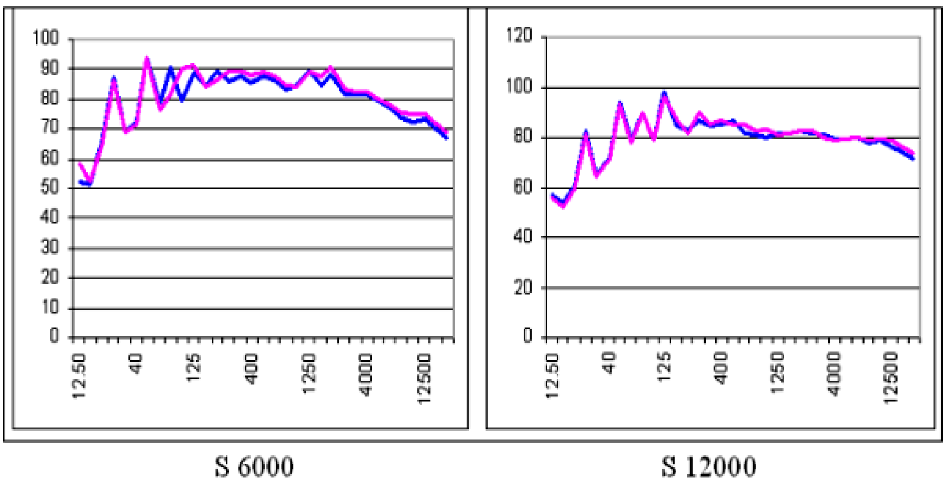


Figure 3. Sound pressure level for S 6000 and S 12000 generator (without and with carter)

The vibration measurements have been performed in order to test the silent block by measuring the three dimensional acceleration values in proximity to the silent block itself. The measures were taken before and after the silent block for evaluating if this was correctly designed. In Figure 4 the velocity value (dB) vs. the frequency of vibration, detected upstream and downstream of the silent block, are compared so as to show the effect of the silent block itself. From this we can observe that the damping due to the presence of the silent block is substantial for high frequencies.

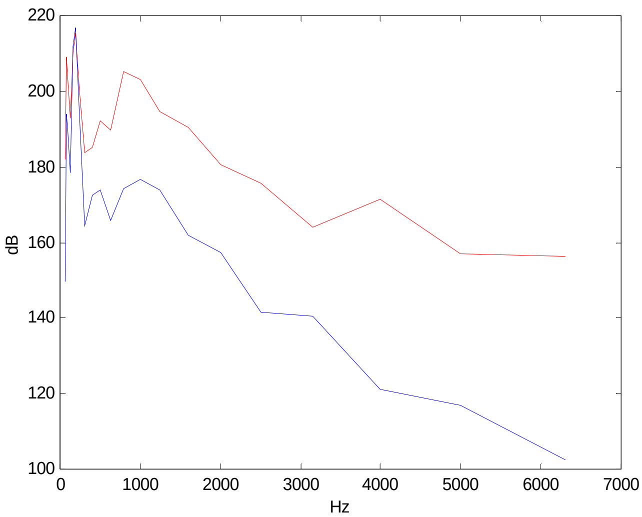


Figure 4. Velocity value (dB) upstream (red) and downstream (blue) of the silent block

As described later, an important task for reducing the noise radiated by the two generating set is to design a new carter; so, after this design phase, the vibration analysis is performed again in order to check the contribution made by the casing to the sound power emission.

3. DESIGN METHODOLOGY

The design of a new range of low sound emission generating set requires the performing of the followings tasks:

- carter and frame design;
- silent block optimization;
- muffler optimization;
- beaded panel design for stiffening the carter;
- others design work.

3.1. Carter and frame design

The partial original casing covering the motor of the two machines allows a very low reduction of sound emission (about 1 dB(A)). Anyway, as stated before, the sound emission from the new generating sets must be almost 2 dB(A) less than the values accepted by the Directive 2000/14/CE so as to satisfy the market targets. This objective has been reached also by developing a new frame and a new casing for the two machines. The results of the preliminary study of the machines are used to design the geometry and the dimensions of the frame and of the casing. The entire lay-out of the machines is re-designed not only taking into consideration the acoustic characterization but also the accessibility for ordinary maintenance, the flow of cooling air and the outlet of hot air into the machine and the need to keep these flows separate and also the ease of assembly.

The adopted design solutions allow for performing of:

- easy realization and consequently low production costs;
- easy panel assembling; the panels are assembled on support clamps with self-locking screws. Between the panel and the clamp it is possible to interpose some damping washers in order to reduce the vibration propagation;
- easy panel realization; these are profiled sheets without the need for welding and a layer of damping material is inserted between the two panels.

In Figure 5 the entire frame and the casing (with the air intake and gas collector) for the S 12000 model is illustrated. The panels are moulded in order to follow the frame tube profile but are not stuck to the frame because of the normal manufacturing tolerances. While the lateral panels are flat and have hot air outlets, the front and back panels are curved (though always simple in form) and with some openings for the cooling air to be transported to the alternator and motor.

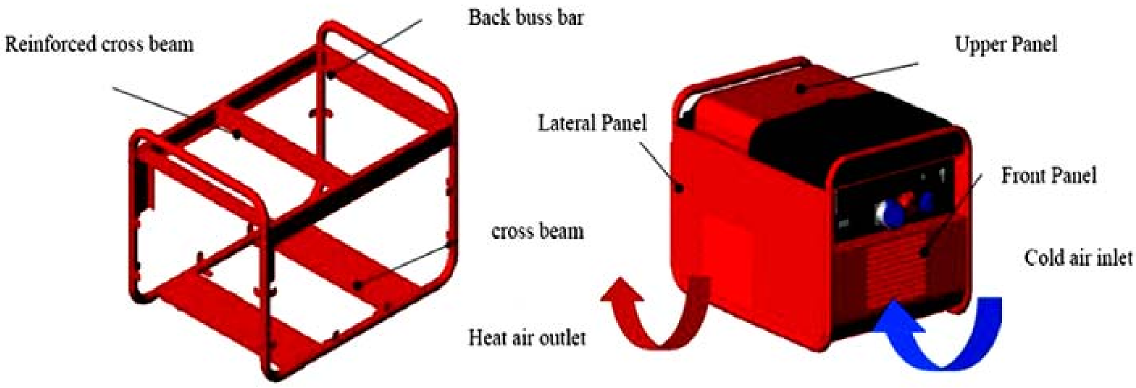


Figure 5. Frame and carter design

3.2. Silent block optimization

The silent block of the two machines were correctly designed and so here this is not studied in detail. This is also because there are other sound sources to consider, not only those caused by the machine vibrations due the motor on the support frame.

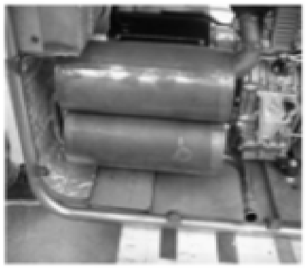
3.3. Muffler optimization

The optimization of the muffler design has been performed considering both the acoustic and fluid dynamic aspects.

In the acoustic optimization another two tasks are performed. In the first one the theory of mufflers [14] is applied in order to obtain a significant noise reduction when compared to the conventional muffler provided by the manufacturer.

Let L_{TL} be the transmission loss factor for a reactive muffler (Figure 6) consisting on two chambers C_1 and C_2 (of length l_1 and l_2 respectively).

The two chambers have a diameter S_1 and S_2 which values depend on the overall size of the muffler (here it is S_1/ S_2 preset to the value 1.2, i.e. $S_1=1.2S_2$) .



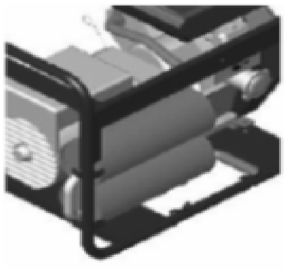


Figure 6.
Two camera muffler

The L_{TL} function for the two cameras (L_{TL1} and L_{TL2}) is given by:

$L_{TL} = L_{TL1} + L_{TL2}$,

where

$$L_{TL1}(f,x) = 10\log\left[1+\frac{1}{4}\left(n-\frac{1}{n}\right)^2\left(\sin kl_1(f,x)\right)^2\right], \tag{1}$$

$$L_{TL2}(f,y) = 10\log\left[1+\frac{1}{4}\left(n-\frac{1}{n}\right)^2\left(\sin kl_2(f,y)\right)^2\right],$$

where x and y are the length of the C_1 and C_2 chamber respectively and $n = S_1/S_2 = 1.2$.

Moreover in equation 1 it is:

$$kl_1(f,x) = \frac{2l_1\pi}{\lambda(f)}x,\ kl_2(f,x) = \frac{2l_2\pi}{\lambda(f)}y, \tag{2}$$

where $\lambda(f)$ is the sound wavelength and f is the frequency (Hz) in correspondence of the internal gas temperature.

The L_{TL} function for the complete muffler is given by

$$L_{TL}(x, y, f) = L_{TL1}(x, f) + L_{TL2}(y, f). \tag{3}$$

Now it is possible to evaluate the mean μ and the variance σ values for $L_{TL}(x, y, f)$ on the frequency domain:

$$\mu(x,y) = \frac{1}{2000\ \text{Hz}} \int_{0\ \text{Hz}}^{2000\ \text{Hz}} L_{TL1}(f,x) + L_{TL2}(f,y)\, \mathrm{d}f\ ,$$

$$\sigma(x,y) = \frac{1}{2000\ \text{Hz}} \int_{0\ \text{Hz}}^{2000\ \text{Hz}} \left(L_{TL1}(f,x) + L_{TL2}(f,y) - \mu(x,y)\right)^2\ \mathrm{d}f\ . \tag{4}$$

For a good muffler design, experimental analysis [15] showed that $\mu(x,y)$ has to be minimized and contemporary $\sigma(x,y)$ has to be maximized. Accordingly, an Objective Function OF(*x*,*y*) is created in order to optimize these two functions:

$$\mathrm{OF}(\mathrm{x},\mathrm{y}) = \frac{\mu(x,y) - \mu_{\min}}{\mu_{\max} - \mu_{\min}} \cdot 1 + \frac{\sigma_{\max} - \sigma(x,y)}{\sigma_{\max} - \sigma_{\min}} \cdot 2 \tag{5}$$

where $\mu_{\max}$, $\mu_{\min}$, $\sigma_{\max}$ and $\sigma_{\min}$ are the minimum and maximum values assumed by, respectively, $\mu(x,y)$ and $\sigma(x,y)$.

The Objective Function OF(*x*,*y*) is showed in Figure 7; the function presents a certain number of local maxima and minima values. The optimization algorithm we implemented is able to detect the global maximum value i.e. the best condition between $\mu(x,y)$ minimization and $\sigma(x,y)$ maximization.

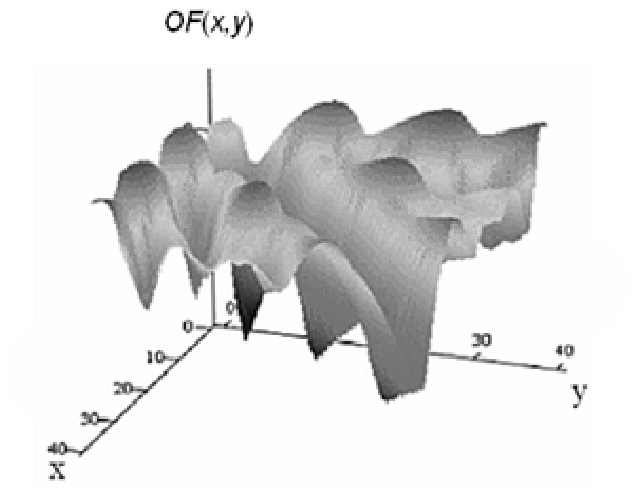


Figure 7. Objective Function

After optimization we have demonstrated that the best compromise between $\mu(x,y)$ minimization and $\sigma(x,y)$ maximization is, for the exposed problem, given by $x = 360$ mm and $y = 240$ mm values.

In the fluid-dynamics optimization we have adopted some innovative design solutions using mufflers provided by hollow pipes and numerous chambers and contractions. In Figure 8 an innovative muffler design, composed by some sub-chambers and perforated separators is proposed.

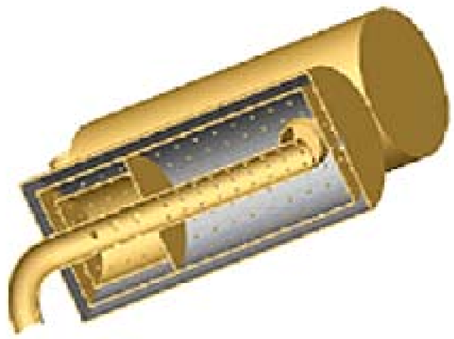


Figure 8. An innovative muffler design

The employed methodology allows for the acoustic muffler optimization. In the case of the Diesel S 6000, for instance, the sound power is reduced by 2 dB(A) using the casing. Since the use of contractions and chambers into the muffler involves the high loss of loads (referring to the gas flow into the ducts of the muffler) a fluid dynamic optimization is performed by using a one-dimensional mean flow analysis. This allows the assessing of the loss of load in a numerical way. This part of the work is to be published in a specialised review.

3.4. Beaded panel design for stiffening the carter

The beaded panel design, useful for stiffening the panels of which the casing is composed, is today under development. Future works will be assessed towards this aspect of research. Moreover the advantages deriving from it, when compared to the other interventions described below, are negligible.

3.5. Others design work

Some others interventions have been undertaken with the aim of reducing the noise of the S 6000 group. The interventions undertaken involve the closure of the motor aspirator that has been positioned on the base of the support frame while the ejection has been modified by inserting a conveyor for the flow. As regards the alternator, the aspiration surfaces foreseen by the design of the casing have been eliminated while the aspirator already present on the crankcase has been amplified. The lining of the casing has been completed with soundproofing material. In total the benefits produced by such interventions are equal to approximately 1 – 1.5 dB(A). Moreover, when more sound power reduction is needed, the use of sound proofing materials is recommended in order to dissipate part of the sound energy inside the motor vane. This, if not absorbed, can increment the level of the sound pressure radiated through the air vents towards the outside. The optimization of the sound proofing material can be achieved through a frequency analysis of the sound energy produced by the generating set when without casing.

CONCLUSIONS

In this paper a general methodology have been described in order to reduce the noise of two industrial machines. In particular some interventions have been identified, at the design phase, for two generating sets, leaving both motor and alternator unaltered.

In this specific case only a few possible interventions have been developed as a sufficient result is still reached (Table 2). The design activity has considered both a new frame and casing design in order to guarantee a passive containment of noise. The design of new mufflers has also been considered.

Table 2. Results

Results obtained for S 6000		***Results obtained for S 12000***	
Configurations	**L_w(A) [dB(A)]**	**Configurations**	**L_w(A) [dB(A)]**
original generating set with casing without casing	 104 105	original generating set with casing without casing	 102 103
generating set with modified casing and original muffler	100	generating set with modified casing and original muffler	98
generating set with modified casing and 2-stages muffler	98	generating set without casing and optimized muffler	97
generating set with modified casing and new muffler with hollow pipes and numerous chambers and contractions	94	generating set with modified casing and optimized muffler	93

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