ICSV14

Cairns • Australia

9-12 July, 2007

14th International Congress on Sound & Vibration

VIBRO-ACOUSTIC CHARACTERISATION OF A D.I. DIESEL ENGINE BY THE USE OF THE SCANNING LASER VIBROMETER TECHNIQUE

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Abstract

The objective of this work is represented by a laser vibrometer application on a diesel engine for the identification of noise sources associated with its structural vibration. The first investigation has been concentrated on the possibility to identify as well as quantify the structural noise emissions of a direct injection diesel engine, used mainly for agricultural application as well as small off-road vehicles.
In fact, the quantity of noise emitted by the structure represents a large part of overall noise level. The structural noise is directly, in fact, associated with the surface velocity of the engine structure and/or of its components and for this reason is very important to correlate the vibration and acoustic analyses.
Through the use of a scanner laser vibrometer (mod. PSV400 Polytec), the vibration maps have been determined of the four accessible investigated surfaces around the engine (according to the standard ISO 3744). In particular, a set of 48 scanning data have been executed: one scanning for each surface of the engine and for 6 different engine speeds in motored and firing operating conditions. The motored condition has been investigated for different engine speeds in order to quantify the mechanical noise associated with the rotating components and finally to compare with the overall noise in the firing condition.
Vibration analysis has been conducted in the investigated frequency range 1 Hz - 4000 Hz. In the last step of the work the vibration data coming from the laser scanning vibrometer have been compared with the acoustic data previously determined through an intensity analysis in order to firstly identify the noise sources and secondly to better understand the nature of the noise, which represents an important information for future optimization.

1. INTRODUCTION

Requirement in designing a quiet, smooth running engine is strictly correlated with the minimization of the engine vibration and noise caused by combustion and mechanical processes.
Internal combustion engine represents a multi-exciting source of vibration and noise

originated by the assemblies and accessories, by whose operation energies are transformed into sound wave energy. The energy absorbed in engine structure excites natural modal oscillations of larger engine parts (cylinder block, oil pump, cylinder head, etc.), through whose surfaces the sound is radiated.
As a part of a more diffuse research program, the aim of this work is to characterize the vibro-acoustic behavior of a small (224 cc) single-cylinder direct-injection diesel engine used for agricultural and industrial applications as well as small off-road vehicles. The activity has been performed through the use of two different experimental techniques (intensity analysis and laser scanning vibrometer analysis) in order to distinguish the air-borne and structure-borne radiated noise. The experimental results are then compared through the evaluation of sound power radiated noise coming from the intensity technique and the vibrometer technique using the analytical formula to calculate the sound power from the vibration level.
Different operative engine conditions in terms of load and speeds were so explored to identify the single noise sources and their weight on the global emission. In particular, determination of sound pressure levels was performed by using the application of Standard ISO 3746 intended for determining the sound power levels of machinery on the basis of sound pressure measurements. The sound intensity analysis by using the Standard ISO 9614 has been used. The vibration analysis in the same engine conditions has been performed, too.
Finally the global results coming from this different techniques have been compared utilizing an analytical formula for the vibro-acoustic correlation. The results have shown the good accuracy of the global data and a good approach for understanding the nature of the noise of such complicated system.

2. TEST ENGINE

The engine is an air cooled, direct injection 69 mm bore and 60 mm stroke representative of a light diesel engine family mainly used for agricultural and industrial applications as well as small off-road vehicles. During the experimental investigations it was mounted on a stationary test bench in Istituto Motori (CNR) laboratory. Detailed specifications are shown in Table 1.

Figure 1. Engine test bench

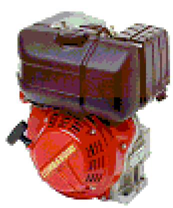
Lombardini 15 LD 225			
	**Cylinder**	**N.**	**1**
	Bore	mm	69
	Stroke	mm	60
	Displacement	cm^3	224
	Dry weight	Kg	28
	Dimension	mm	265x158x417

Table 1. Engine characteristics

For determining the level of emitted sound in individual areas of the engine at various speeds of running, detailed researchers were carried out by using experimental vibration and sound

measurements, both in pressure, intensity and vibration field. In that manner, it was characterized the noise source under steady state conditions, both under motored and firing conditions. The results was then compared.
A schematic drawing of the engine test bench is shown in figure 1. According to its functional behavior, it includes the following modules: the diesel engine, an electrical dynamometer, the fuel injection line, the data acquisition and control units as well as the emission measurement system. The electrical dynamometer allowed to operate both in motoring and firing conditions that was appropriate to detect the in-cylinder pressure data and to explore the engine behavior in stationary and simple dynamic conditions. The pressure regulator valve and the solenoid injector were driven by an in-house electronic system.

3. NOISE MEASUREMENTS AND ANALYSIS

In order to measure engine noise with reasonable accuracy the acoustic characteristics of the measurement environment must be known. On an acoustic basis, the ideal environment is a space with no reflecting surfaces and no background noise. In practical terms the 'best' environment, for diesel engine applications, is an open-air site with one hard reflecting surface (the ground), no other obstructions for at least 50 m from the noise source and all the microphone positions, and background noise levels which are at least 10 dB (and preferably 20 dB) below those to be measured. Alternatively, engine noise measurements could be made in 'non-anechoic' test cells using a standard test cell with some acoustic absorption (like the used laboratory for the experimental validation). For tests involving sound pressure measurements, this cell have to be calibrated by the measurement of the reverberation time: **ISO 3746**. Alternatively, sound power levels can be obtained using acoustic intensity analysis equipment where no particular test cell is required: **ISO 9614**. Both two approaches has been used to characterize the source and to determine the engine radiated noise.

3.1 Sound pressure method

In order to detect the mechanical noise a preliminary sound pressure level measurement was made under motored conditions to determine the frequency range of significant noise emission in five different engine speeds (1400, 2000, 2500, 3000, 3400 rpm).
The method described in ISO 3746 is used for "in situ" conditions. The A-weighted sound power level of engines is determined on the basis of sound pressure measurements carried in an open space or room over reflecting plane.
The measurement plane, (where measurement points are located), should be located in area surrounding engine over reflecting plane. A rectangular surface was used, see fig.2. Measurement distance, according to the method (the perpendicular distance between reference rectangular and measurement surface) should not be lower than 0.15 m, and is suggested to be equal to 1 m.
Measurements were carried with the use of a class 1 precision devise. The minimal number of measurement points equal to 5 was used to meet requirements set in Standard. This method has permitted the identification of critical frequency ranges and related noise sources in the different operating conditions.

A typical sound power spectra for the motored condition at 2000 rpm engine speed is reported below.

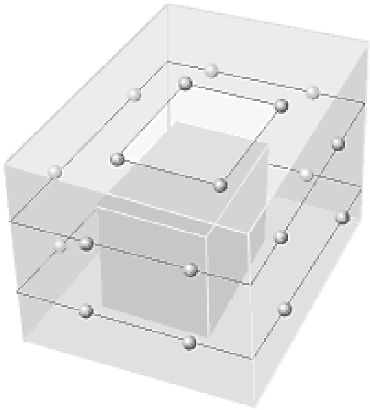


Figure 2. ISO-3746 rectangular surface

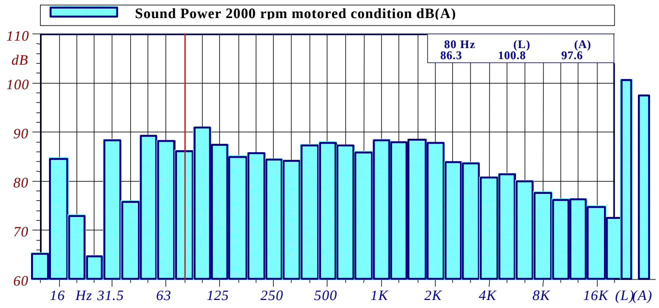


Figure3. Sound Power dB(A) at 2000 rpm-motored condition

3.2 Intensity analysis

Sound intensity measurements are quite useful as an acoustic measuring technique, since in a reverberant environment the free-field as well as the diffuse-sound field are determined.
For the sound analysis a Bruel & Kjaer intensity probe has been used with a spacer of 12 mm (available frequency range 125<f<5000Hz, f=frequency). The probe was connected to a Leuven Measurement Systems (LMS) CADA-X FFT-based measurement and analysis system. The intensity and sound pressure level were measured simultaneously. The sound power level was measured around a rectangular surface divided in 70 sub-areas see fig.4, in two different cases (motored and firing conditions) and for different load (1000, 1500, 2000, 2500, 3000, 3400 rpm) conditions.

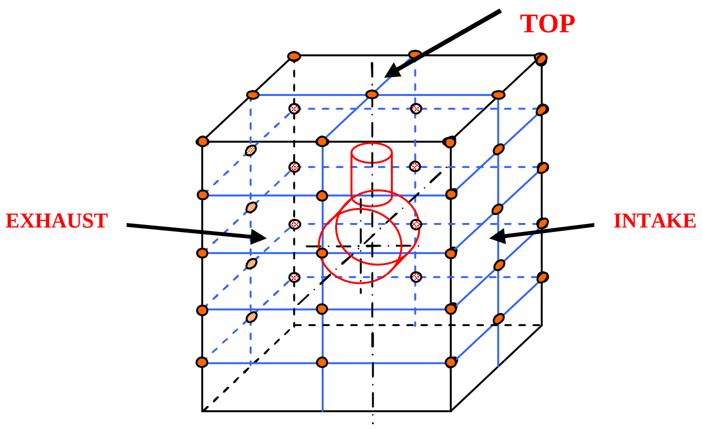


Figure 4. Surface of intensity measurements according to ISO 9614

The engine test cell is a room of 5 m x 4 m and with a 4 m of high, with an absorption treatment to the wall and to the ceiling.
For a detailed investigation of sound radiation the point method as described by **ISO-9614** part 1, was used and a contour plot of the intensity vector was utilized.
The results of the sound intensity analysis have the advantage that they are based on spatial averages, hence they are highly representative and not local in occurrence. On the other hand, since sound intensity is measured at a certain distance from the engine surface, the results cannot always be clearly assigned to a certain engine component. Therefore, the interpretation and understanding of sound intensity results need to be supported by the results of surface vibration analysis which are directly related to engine surface parts and their locations

(reported in the next paragraph). In the table 2 is reported the surface acquisition points around the engine:

Surface: 4x4=16 points for each surface
Total surface: 5 (Top, Front, Exhaust, Intake, Back)
Total points acquisition: 80 around the virtual suface (parallelepiped)
Operating conditions: 1000,1500,2000,2500,3000,3400 rpm (firing and motored conditions)
TOTAL ACQUISITION : **960** experimental measurements for the intensity analysis

Table 2. Experimental acquisition points

In fig. 5 is shown the experimental layout measurement, where is visible the grid points and the used sound intensity probe during the experiments.
In Figures 6-7 are reported the intensity results, respectively, a sound power contour map (dB(A)) at 3000 rpm and the comparison of sound power dB(A) at 1500 rpm both for firing and motored condition measured at the 5 used surfaces.

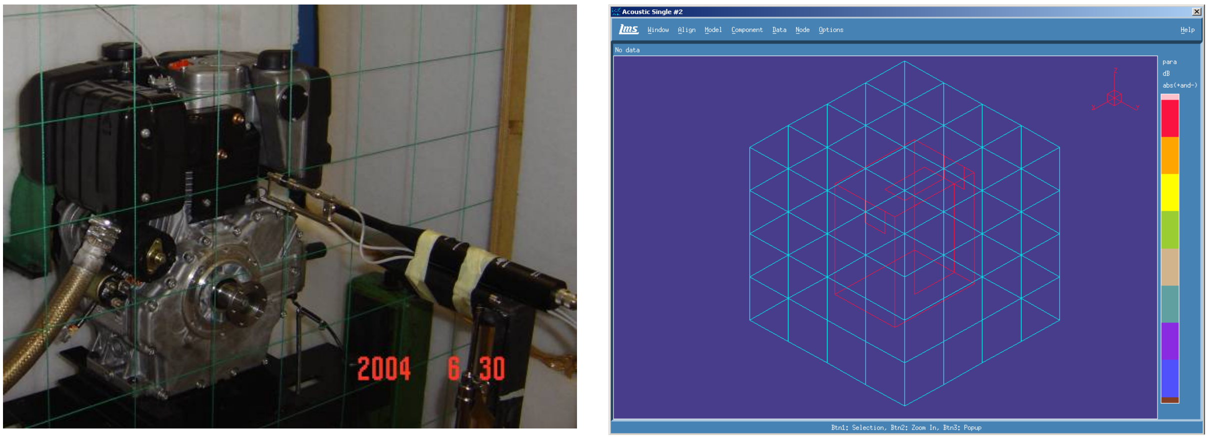


Figure 5. Intensity measurement and surface field points

By viewing the figures 6-7 it comes evident the different contribution of the single component of the engine to the overall noise radiation. The mayor noise source contribution comes from the top surface that corresponding to the combustion process, due to the rapid rising rate of cylinder pressure. The obtained acoustic results were then compared with the vibration data to better drive a future optimization process.

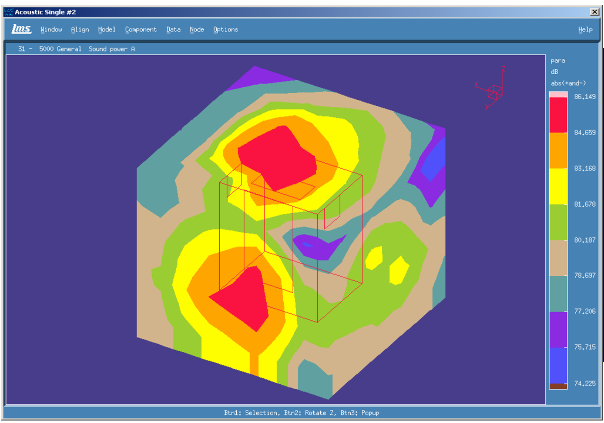


Figure 6. Sound power contour map (dB(A))

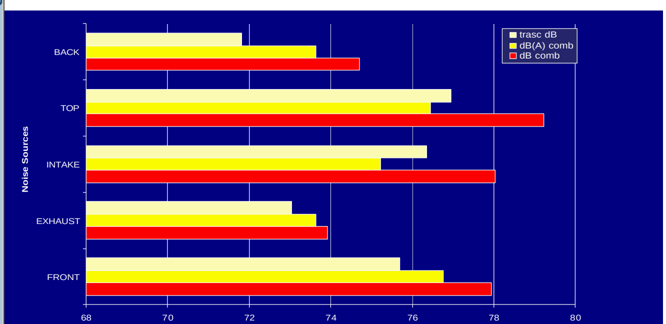


Figure 7. Sound power comparison in the two different operative conditions
Red bar: ____sound power dB
Yellow light bar: _____ motored condition

4. VIBRATION ANALYSIS

The final experimental setup consisted of a scanning laser vibrometer system. Such system provideed the ease of acquiring the vibrational response for a large number of non-contact measurements around the tested engine.
The scanning head was elevated in order to be normal to the testing engine surface. This instance was to obtain better reflectivity and hence better Signal-to-Noise ratio (SNR).
The experimental campaign was executed in the same condition of the acoustic analysis and the results compared. Following (figure 8) are reported the different mesh used for only two measured surface.

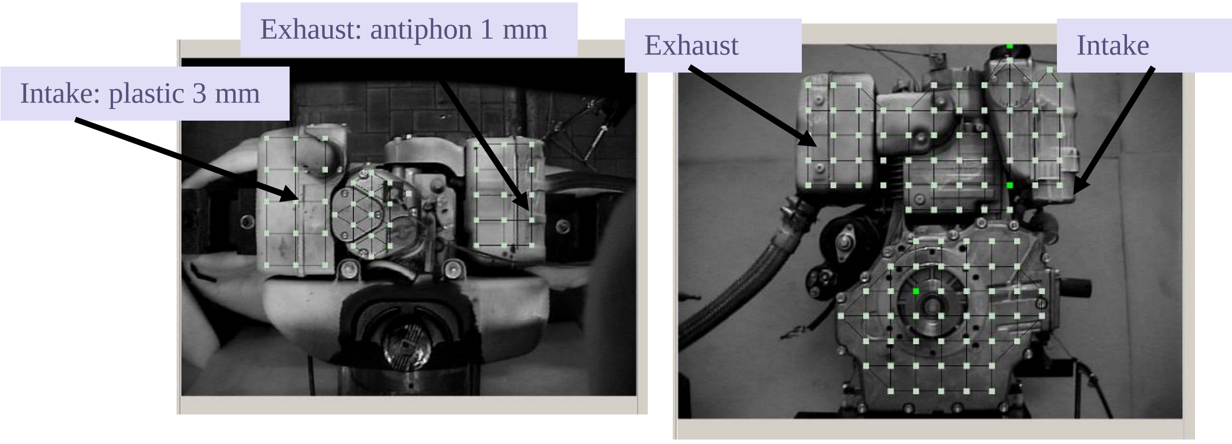


Figure 8. Vibrational mesh (left: top surface, right: front surface)

In figure 9 is reported a typical velocity frequency spectrum at 1000, 2000, 2500 rpm where is possible to note how the peaks change with the frequency in correspondence of its fundamental frequencies. The three circled areas in the figure represent zones of amplification of the fundamental peaks independently of the harmonic order.
It is possible to affirm that a possible resonance phenomena is present in correspondence of that frequencies.

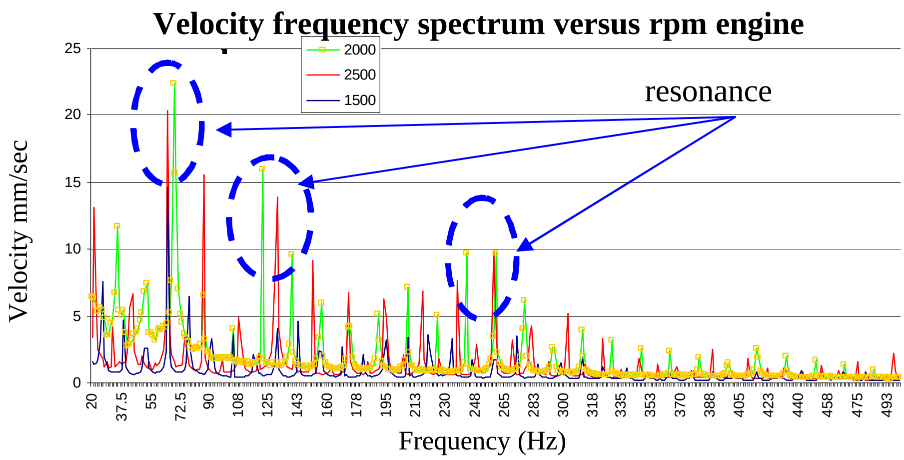


Figure 9. Velocity frequency spectrum vs rpm

Next figure 10 shows a typical velocity distribution on the front side of the engine, too.

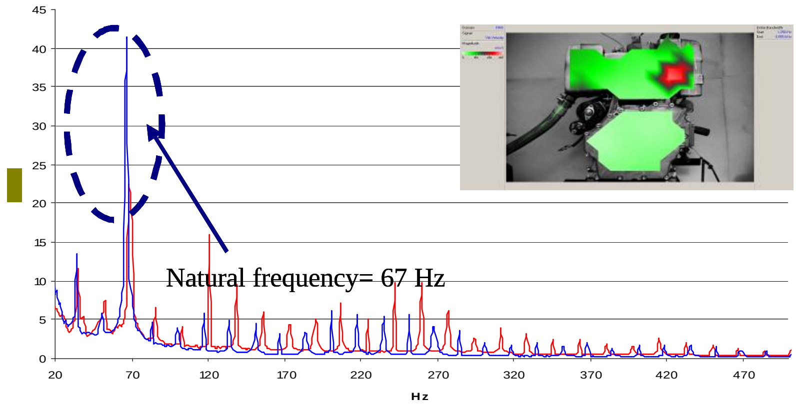


Figure 10. Velocity frequency spectrum at 2000 rpm (front surface:114 acquired points)
Red line: firing condition
Blue line: motored condition

2.1 Analytical vibro-acoustic correlation

As mentioned before, the vibrational data may be utilized to evaluate the sound power radiated because of the close correlation between the body engine vibrational levels and the acoustic emissions. The equation 1 express the sound power directly proportional to the rout mean square velocity:

$$W_{rad} = \rho \cdot a \cdot S \cdot < v^2 > \cdot \sigma_{rad} \tag{1}$$

where:

$$\lambda_a = \frac{2\pi}{a} \cdot \frac{\sqrt{\frac{E}{\rho}}}{\sqrt{12 \cdot (1-\nu^2)}} \cdot h \qquad \sigma_{rad} = \frac{\lambda_a^2}{S} \tag{2}$$

substituting equations 2 in equation 1:

$$W_{rad} = \rho \cdot a \cdot < v^2 > \cdot \lambda_a^2 \tag{3}$$

where: ρ = air density; a= sound velocity; σ_{rad} =acoustic radiation efficiency; h= panel thickness; S= radiated surface; <v2>= velocity root mean square; λa = sound wave length.
It is important to observe that the radiated acoustic power W_{rad} depends on the frequency because of both her < v2 > that σ_{rad} are dependent from the frequency.
The parameter that plays an important role in the sound power evaluation, is σ_{rad} that depends from the geometry and from the material property of the structure.
Particularly, considering a panel of assigned surface, the radiation efficiency will vary with the material and the thickness of the panel.
Finally, using previous formula starting from vibrational data, the sound power radiation has been evaluated and compared with coming from the intensity analysis, in order to better understand the nature of the noise and to thinking about a future optimization system.
In figure 11 is reported the vibro-acoustic comparison, performed for each measured surface (in total five surfaces).

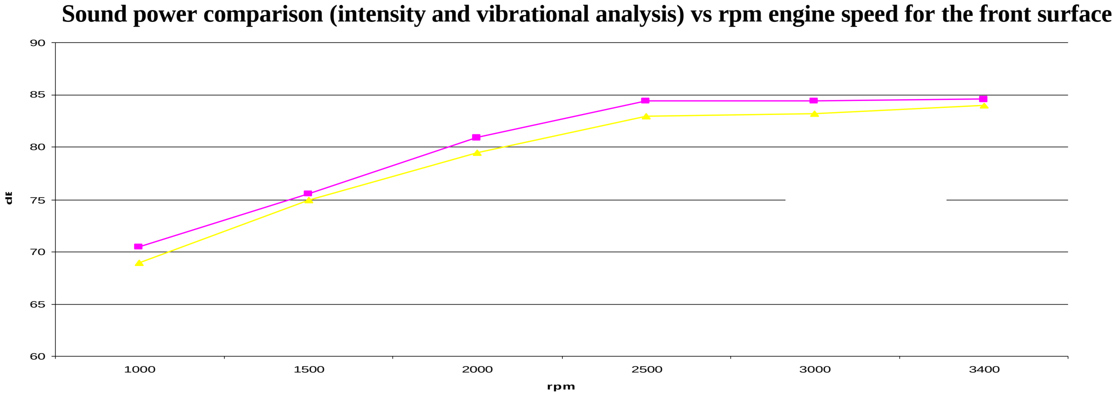


Figure 11. Sound power comparison vs rpm (Front surface)
Pink line: intensity analysis
Yellow line: vibrational analysis

The previous figure shows the correlation accuracy and the good according of the data. Only a 1 dB of difference is appreciated between the two used analyses. The sound power difference, as expected, depends on the fact that the vibrometer represents an instrument to evaluate the structural share.

CONCLUSION

From the vibrational analysis and with the aid of dedicated analytical formulations, it has been possible to calculate the acoustic power associated to the mechanical noise.
Such data have been compared with the data obtained through the intensity analysis.
From the comparison emerges a good accuracy of the analytical formulation that can represent an important tool in the following phase of study to evaluate corrective solutions finalized to the noise reduction.
Activity is still in progress and also will foresee the use of numerical simulation (FEM-Aero-acoustic).

ACKNOWLEDGEMENTS

The author would like to express her acknowledgement to Dr. Viscardi Massimo for the valuable technical and analysis support during the experimental analysis.

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