Machine Vibration (1996) 5:63–70
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Machine Vibration

Noise and Vibration Analysis of an Automobile Engine Component

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This paper describes a mixed experimental/numerical analysis of the dynamic behaviour of an automobile engine part. The experimental analysis involved the use of an optical interferometric technique, Electronic Speckle Pattern Interferometry (ESPI) and a discrete modal procedure with accelerometers. The numerical investigation was performed with the help of a ADINA Finite Element (FE) package. The ESPI is a very accurate non contact field technique that allows a preliminary modal analysis (Jones and Wykes [1]). Thereafter all points of the foregoing investigation can be selected, thus reducing the amount of data to be processed in the discrete procedure with accelerometers.

Finally, a measurement of the acoustic pressure and intensity with the component fitted to the engine in the car led to the acquisition of the sound spectra. These data allow identification of the noise emission from the part under study, and its contribution to the global engine noise.

Keywords: Acoustics; Engine component vibrations; ESPI; Modal analysis

1. Introduction

The noise emission from an automobile engine is a very important design factor, as it impacts not only driver and passenger comfort, but also the external environment. But weight savings, to reduce fuel consumption, require the automobile parts to be as light as possible. These parts, for example body and thin-walled steel pressed engine components, are also the most important noise emission sources. Noise reduction may be achieved by using sound absorption lining materials, with unavoidable weight increase, or by adopting a better design of such parts.

The experimental approach reveals, in a very reliable way, the zones of largest vibration amplitude, suggesting where stiffening ribs may be included, and the improvements to be obtained in vibration amplitude reduction. As the engine is the main excitation source, the aim of the design optimization is to shift the eigenfrequencies of the component under analysis to a favourable zone of the vibration spectrum induced by the engine.

In this work the modal vibration analysis of an automobile engine chain drive distribution cover is performed. The engine cover was tested with accelerometers for a discrete experimental analysis and with time average ESPI techniques for a field analysis.

2. Vibration Modes Assessment by a Qualitative Field Technique

In a first step, an interferometric optical technique was used for determining the eigenshapes. Being a non-contact field technique, there is no need to apply additional masses to the structure, as would result from the mounting of transducers. Thus, the obtained information is continuous and complete, revealing the full behaviour of the vibrating body. Stationary and maximum amplitude points are well-identified, suggesting the relevant points for a quantitative analysis to be described in the next step. Figure 1 shows the ESPI set-up that was used for this experiment.

The mechanical excitation of the component may result from sound coupling (using a loudspeaker, for example) or by using special low-mass vibrating devices. Due to the high resolution of ESPI (of the order of the laser emission wavelength) it is possible to make measurements at very low excitation energies. The mechanical coupling that results from sound transmission through the air decreases dramatically as the frequency increases; thus, when one wants to study

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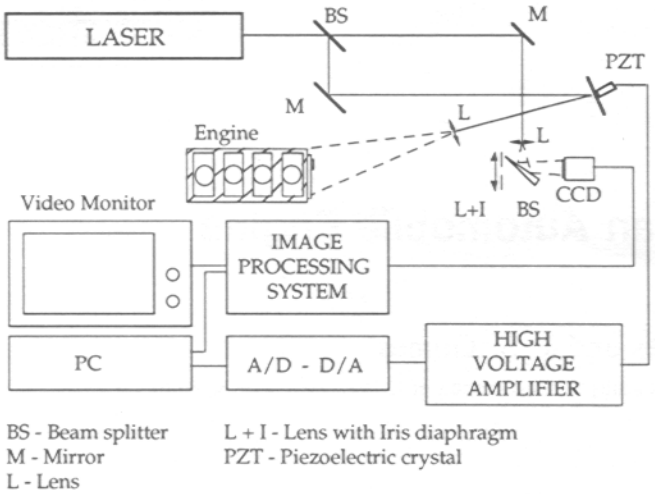


Fig. 1. ESPI set-up for time-average measurement of the eigenmode shapes of the cover plate.

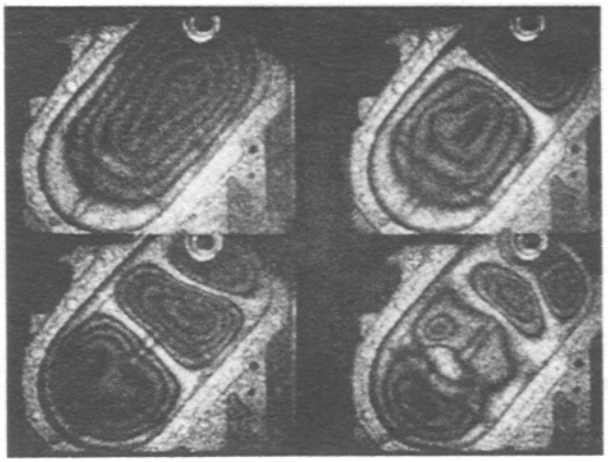


Fig. 2. The ESPI fringe patterns referred to the first four vibrating modes of the cover plate.

high frequency vibration modes, these cannot be induced by sound transmission, and use of PZT devices, smaller than a postage stamp, which can be bonded to the structure is a better alternative. In this case, the test part was excited in pure harmonic vibration with the help of a PZT ($12 \times 3 \times 0.5$ mm; 0.5 g, from Vernitron Limited) bonded on a zone of the surface cover where the studied modes of vibration should not present stationary nodes.

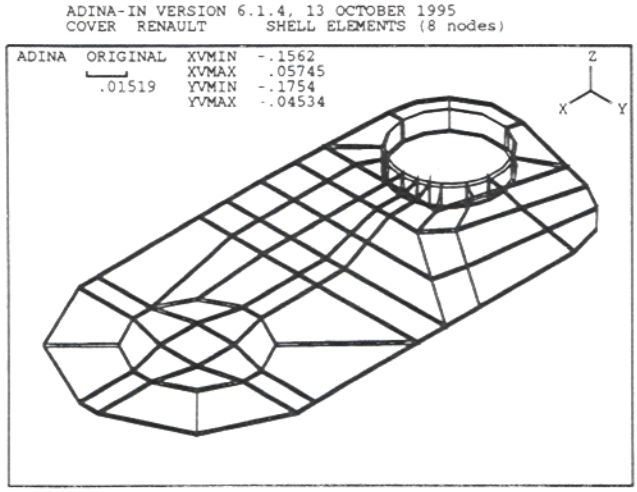
Prior to the component excitation, the surface is illuminated with the laser light and the resulting speckle pattern is recorded in the computer memory. This image is real time subtracted from the incoming video images recorded with the object in harmonic vibration. A frequency sweep is performed so that the lowest frequency eigenmodes which would be excited by the engine, and which are undesirable to the human ear are detected. Figure 2 shows the first four modal shapes, as revealed by the ESPI technique. In the ESPI analysis, the use of a time-average procedure leads to fringe patterns where each fringe refers to points of equal transverse vibration amplitude. Each picture shown in Fig. 2 is the result of the integration of several cycles of vibration; thus the intensity of the fringe pattern is modulated by a J_0 Bessel function. Use of a phase modulation device, consisting of a mirror mounted on a PZT transducer, allows determination of the relative phase between the several *bulls eye* shapes. In this phase modulation technique, the mirror is put in vibration with a frequency slightly different from the object vibration, leading to a beat phenomenon, which gives the illusion of a slow motion vibration (Løkberg and Slettemoen [2]).

Fringe visibility can be increased by using as subtraction reference, an image where only the auto correlation terms were included. This image can be recorded by setting the phase modulation mirror in motion with an amplitude corresponding to that of a zero of J_0 (zero order Bessel function) [3]. The resultant image has a good contrast; however, speckle noise is still present. The image quality can be improved by using speckle noise reduction algorithms by which the final fringe pattern is the average of several speckle decorrelated patterns [2]. The illumination and observation directions were chosen in order to have the sensitivity vector oriented normal to the surface of the object.

3. Numerical Analysis

In parallel with the experimental work, a numerical analysis of the part with ADINA eight-node shell finite elements (FE) and consistent mass distribution [4] was performed. CAD (Computer Aided Design) data were used for prior definition of the FE modelling. The mesh was adjusted in its geometry and refinements were considered where the ESPI images revealed highest fringe concentration. In the geometry modelling, some simplifications had to be allowed due to the high number of degrees of freedom (DOF), which means high computation costs. The seal housing collar introduced some difficulties in the FE solution calculations, possibly due to the high membrane stiffness of such a collar compared with the bending deformation of the adjacent plate elements. In the optimized final mesh this detail was suppressed, leading to a better agreement between the numerical solution and the experimental ESPI results. Figure 3 shows two of the meshes used.

Concerning the boundary conditions, the side web of the cover (See Fig. 4), was considered in a first



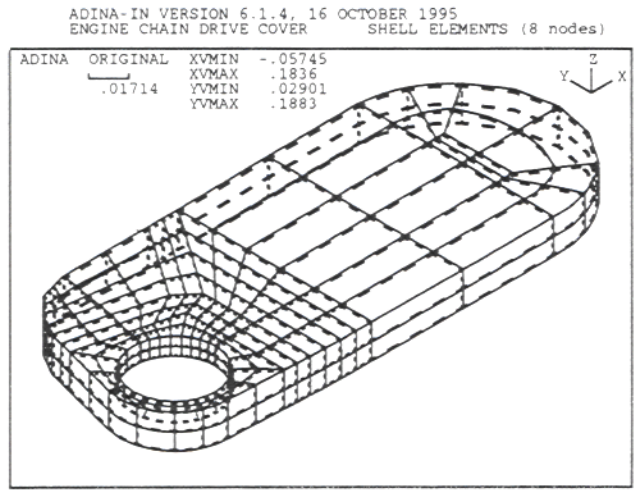


Fig. 3. (a–b). Two of the meshes used in the numerical modal analysis.

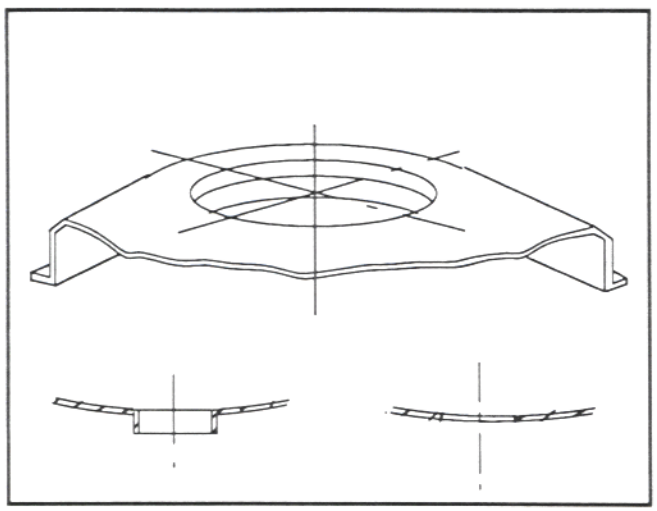


Fig. 4. Estimated deformation of the cover close to the sealhousing with and without collar stiffener.

solution with the FEs; however, this led not only to a high number of DOF but also to a less accurate solution when compared with ESPI results. This may be due to a less realistic modelling of the geometry of the object which, in reality, has no sharp angles. The suppression of the side web proved to be a good alternative for the final FE approach, as a good agreement can be observed between the numerical and experimental analysis, at least for the first four modes of vibration, as shown in Fig. 5.

At this step (the optimized mesh geometry), one can improve the accuracy simply by choosing the numerical integration order in the elements (low order integration means a more flexible element). The results were post-processed so that a set of equal amplitude vibration, (iso-curves), in the same direction of the ESPI sensitivity vector could be represented.

4. Discrete Modal Analysis

This step consisted of an experimental modal analysis using discrete techniques with accelerometers. Frequency response function measurements [5], were achieved by accelerometers mounted on selected points of the cover plate, with post-processing of data with a spectral analyser. The mesh showing the nodal points where measurements were taken is shown in Fig. 6.

From this set of points it is possible to assess the frequency response functions, either by exciting a single point and measuring all the others or, conversely, by changing the excitation point and measuring the response at a single position. For the first procedure, a white noise driven shaker was used as an excitation source. In the second procedure, this was replaced by a B&K instrumented hammer. In all cases, the vibration transducer was an accelerometer. Both the signal of the excitation and the response were fed into a spectrum analyser. The recorded data were post-processed to identify the eigenmodes using the LMS CADA-PC package. The graphical animation of the identified eigenmodes reveals a good agreement with the eigenshapes identified with the ESPI system and FE ADINA.

The experimental data were obtained from measurements with the cover bolted to the engine block at the recommended torque and the engine block laid on a bench. For the second case the engine was tested in the car. The excitation consisted of running the engine at idle and normal regime speed (respectively 550 and 3000 rpm). Fig. 7 presents the amplitude *vs* frequency results. The peaks in the amplitude reveal the eigenfrequencies and it is interesting to note that these values are in good agreement with those previously obtained, at least for the first four eigenvalues.

In the second step of the experiment the cover was bolted to the engine block fitted in the car, which was then started up. The aim of this test was to check

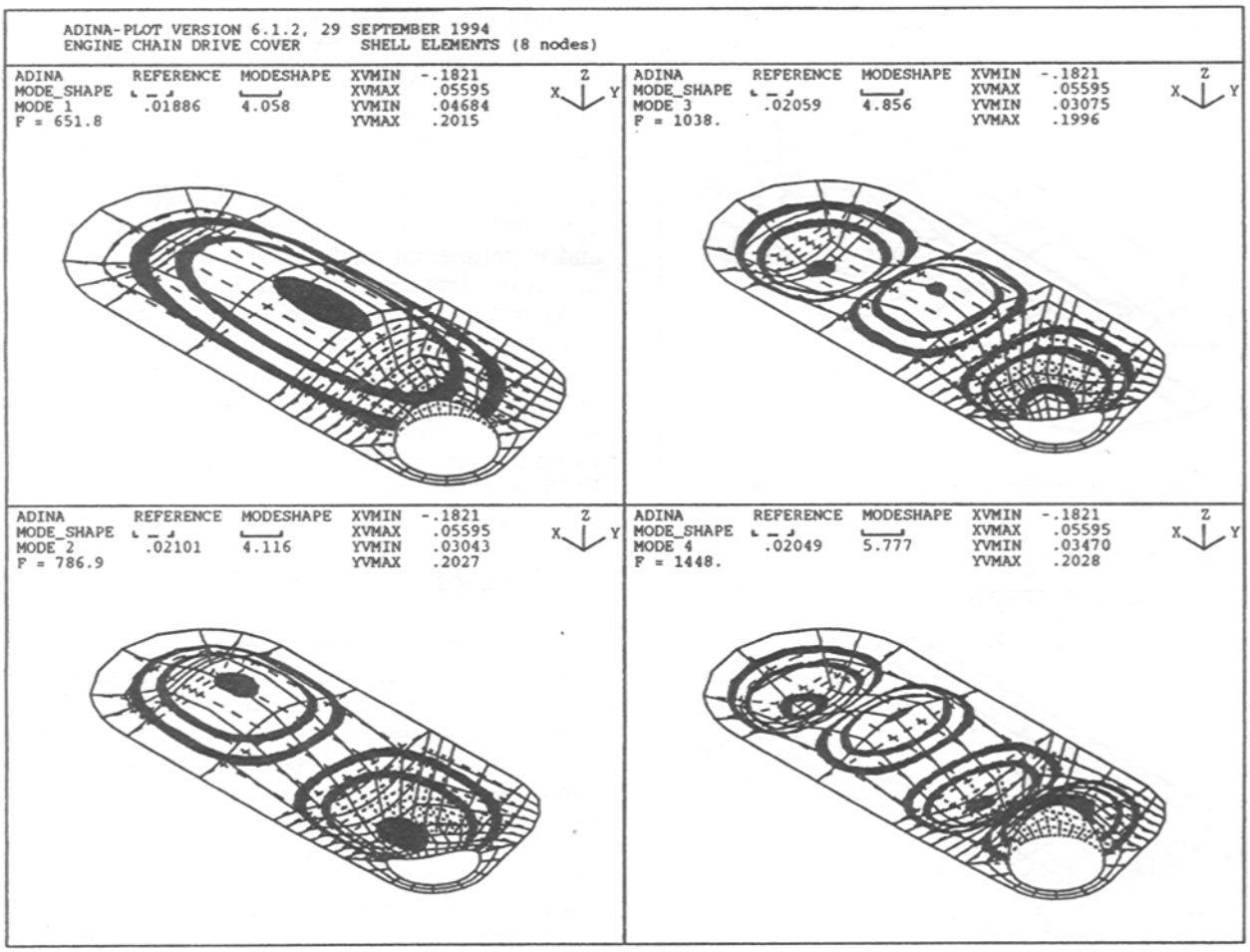


Fig. 5. Mode shapes of ADINA analysis.

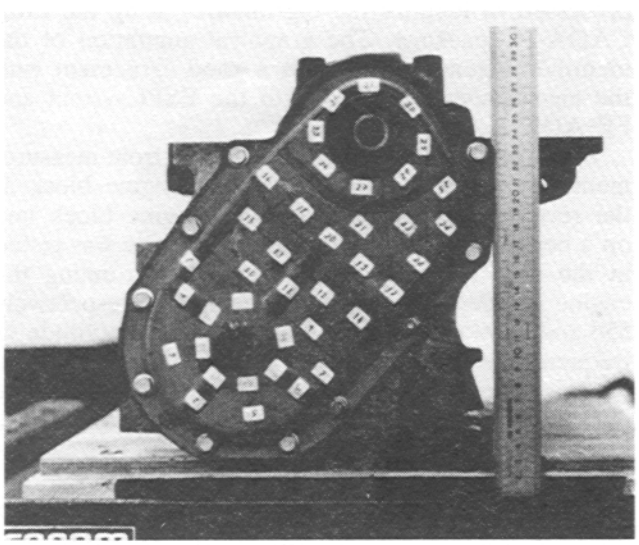


Fig. 6. Measurement point mesh.

which of the eigenmodes were excited when the engine is running normally. To do this, an accelerometer was placed successively on several points of the same mesh used in the previous analysis (Fig. 6). The running tests were performed at 550 rpm (low idle) and 3000 rpm, and the results presented in Fig. 8 reveal that here also the eigenmodes of lowest frequency are excited. From these results, it can be seen that when the acceleration measurements are taken on a nodal point of certain eigenmodes, there is an important attenuation in the amplitude, as expected. It is noticed that on point 19 the second mode does not appear.

To check whether the obtained results really correspond to the contribution of the cover, an accelerometer was bonded on to a point of the engine block and oriented along the same direction as in the cover as shown in Fig. 8.

5. Measurement of Pressure and Intensity of Sound Spectra

The aim of this analysis was to check the contribution of the cover vibration to the global noise emission

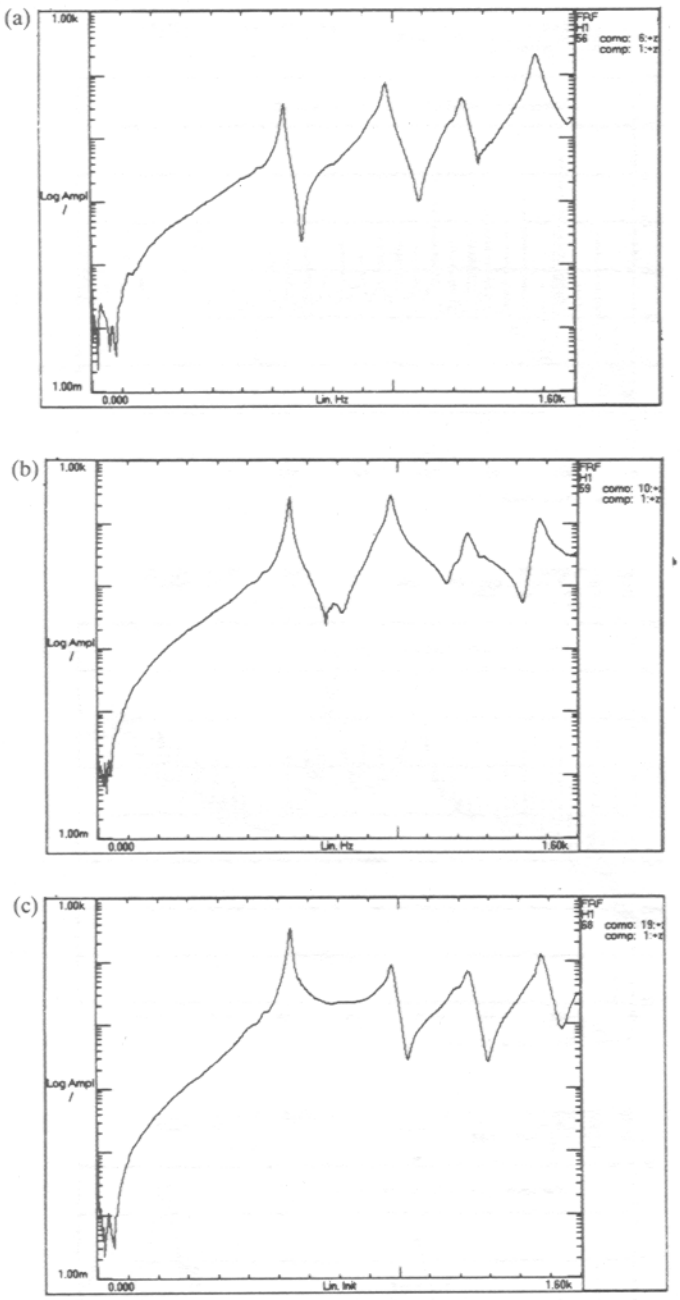


Fig. 7. (a–c). Frequency response functions for points 6, 10 and 19 respectively.

spectrum of the engine. To do this, the car was placed in a semi-anechoic room as shown in Fig. 9. An accurate speed control was implemented with the help of a stroboscopic flash and measurements for sound pressure and intensity were recorded. For the acoustic pressure measurements a microphone was placed over the engine on the same side as the chain cover. For the sound intensity measurements, a six coarse mesh with six sub-areas was defined over the engine with the total area of 1.05 m^2.

Figure 10 represents graphically the spectral intensity of the sound measurements. It is noted that when the engine runs at idle the cooling fan is the noisiest part, but at higher speeds it is the cover side that gives the maximum contribution to engine noise.

Sound intensimetry proved that when the engine runs at low idle, the noisiest area corresponds to the cooling fan zone, (normally on, because the car is stationary and no normal cooling air flow was available), whilst at fast running it is the cover zone that makes the greatest contribution to total noise emission.

Table 1.

ESPI	ADINA	Error (%)	Modal analysis	Error (%)
651	651.8	0.12	639.8	1.72
847	786.9	7.1	978.4	15.51
1530	1038	32.16	1220	20.26
2237	1448	35.27	1470	34.29

6. Conclusions

As can be seen in Table 1, the three techniques are in quite good agreement for the first two natural frequencies and the error increases with the eigenfrequencies. The higher eigenmodes are however associated with a larger vibration energy and so, they have smaller amplitudes, which makes an insignificant contribution to the global noise emission. This was evidenced in the noise spectral analysis.

These experimental tools proved useful in the design improvement of the assessed component, as they are complementary to a previous numerical analysis made with FEM (Finite Element Method). The joint procedure made with the experimental and numerical techniques allows for a more realistic meshing Steinvender [6], which can then be used for testing new design solutions with the optimized numerical model. Therefore, new design solutions can be tested using only numerical procedures.

Acknowledgments

The quantitative measurements were performed in the Laboratory for Acoustics and Vibrations of IDIT (Institute for Development of Industrial Technology in North Portugal). The authors want to thank their colleague J. Fonseca for the facilities in using his old car that proved to be a very good noise source, and the technical staff of the Laboratório de Vibrações do IDIT, particularly the colleagues Santos Pereira and Arménio Vale for helping with the acoustic and vibration measurements.

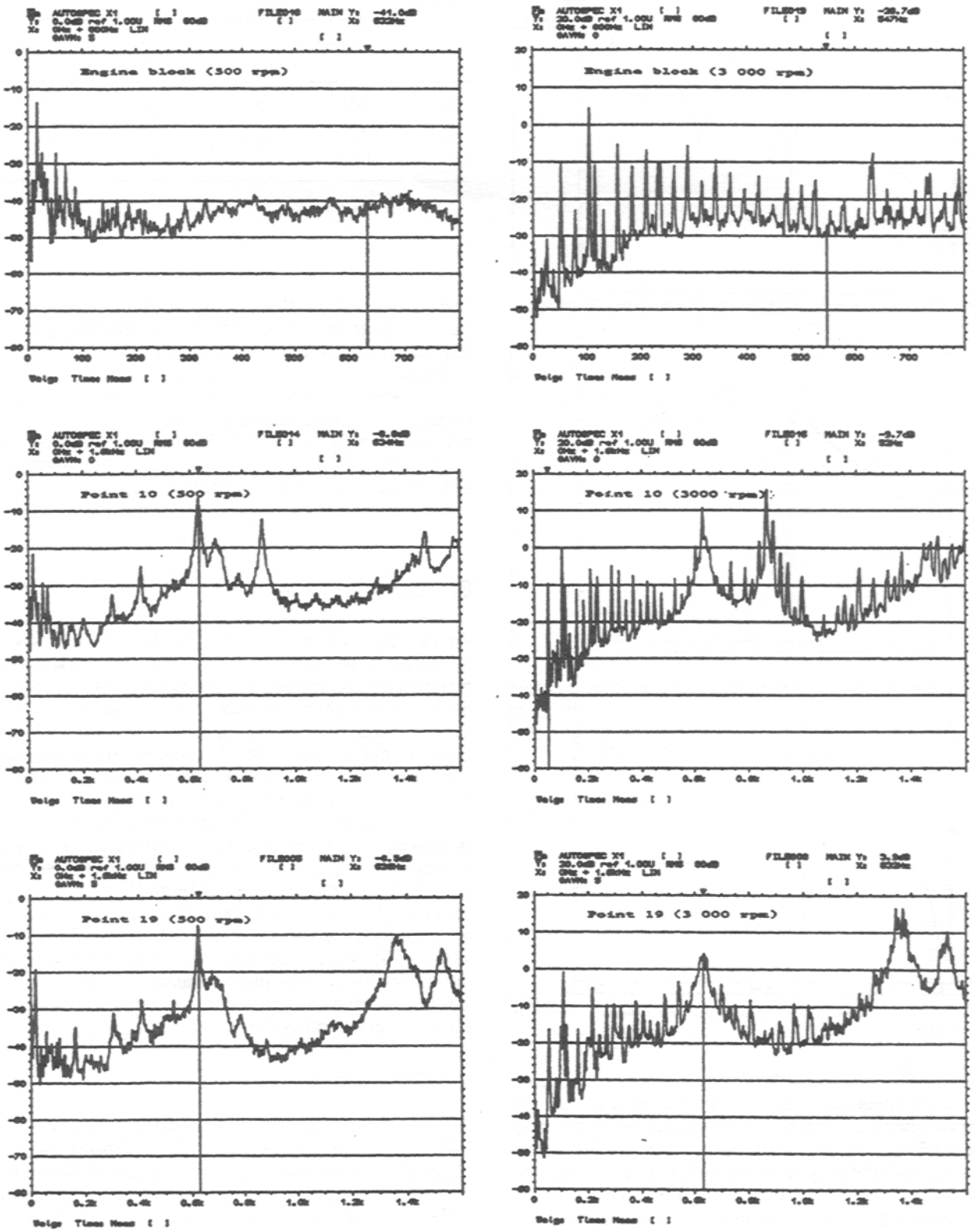


Fig. 8. Auto spectra for several points on the cover and on the engine block; (excitation results with the engine running).

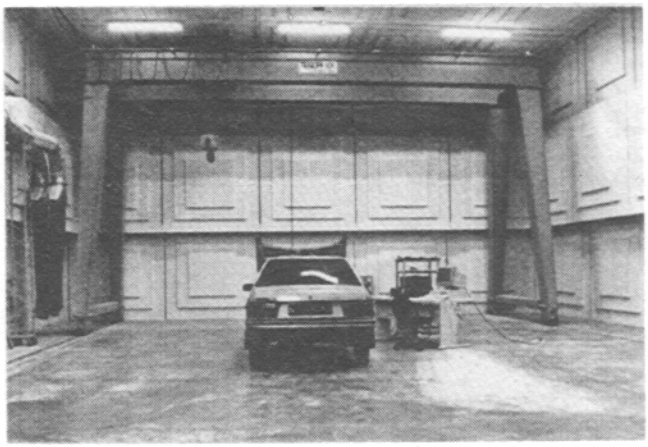
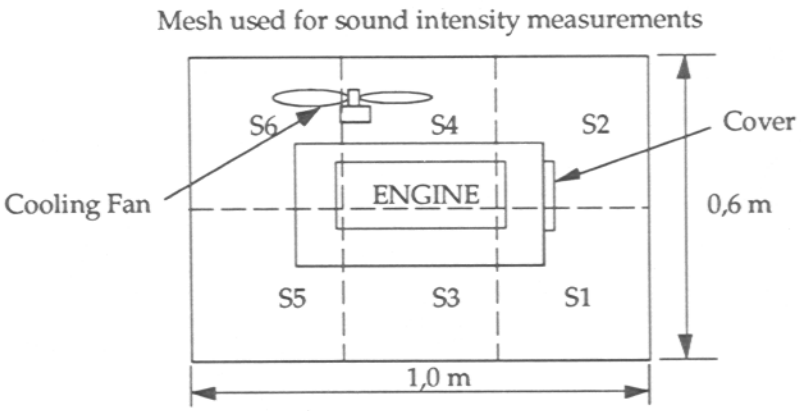
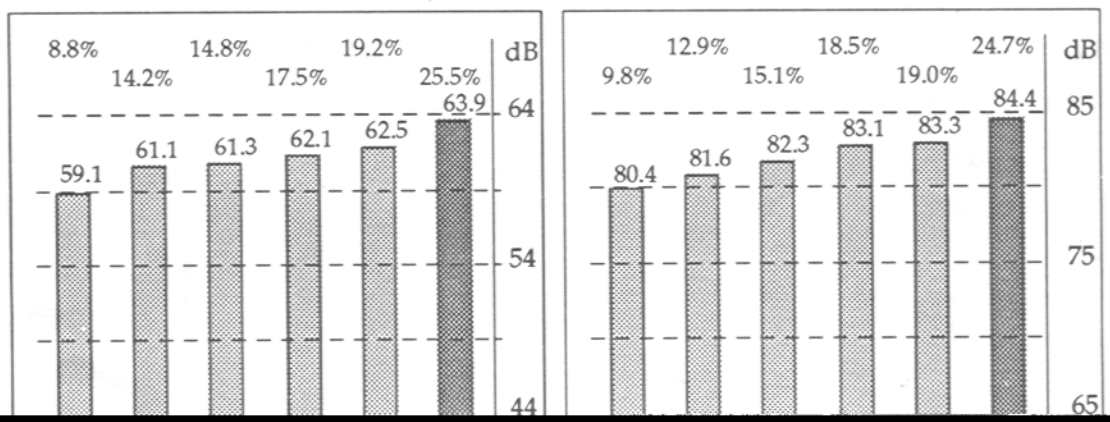



Fig. 9. Car tested inside the semi-anechoic room.