CONSTRUCTION MACHINERY CAB VIBRO-ACOUSTIC ANALYSIS AND OPTIMISATION

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SUMMARY

Construction machinery manufacturers must comply with strict European legislation requirements in order to guarantee operator safety conditions. In this respect, noise reduction and vibro-acoustic comfort are factors of ever-growing importance in the machine design and great effort is focused to their improvement.

In addition, simulation has become usual practice aimed at solving problems associated with vibro-acoustic control for several categories of products ([1] to [5]). This approach is indeed less expensive and time consuming in comparison with the experimental one. It does not need the actual implementation of noise treatments and carrying on extensive field measurements. Moreover, it allows the prediction of noise fields during the early design stage of product development.

A research project has been undertaken at IMAMOTER Institute in collaboration with the Department of Mechanical Engineering of the University of Trieste, to investigate effectiveness and perspective of such an approach in construction machinery field. Its final goal is the identification of solutions which optimise the cab vibro-acoustic performance. The present paper describes the results achieved in the early phase of a parametric analysis aimed at simulating the vibro-acoustic field at a wheel loader operator head location.

A 3D cavity representing the earth-moving machine cab has been modelled by means of a FE structural mesh (Ansys), reproducing the characteristics of the real structure. Starting from the cab vibration load experimental acquisition, a BEM coupled analysis (Sysnoise) has been carried out to evaluate the cab inner vibro-acoustic field as a function of the physical properties of each structural element. A multi-objective design optimisation code (modeFrontier) drives the analysis process flow taking into account the cab parameter structural modifications and carrying out the vibro-acoustic field optimisation.

In this study it has been assumed that the generation of the interior sound pressure field derives from structure-borne noise only, since this is prevailing on the air-borne noise at the frequencies of interest.

1. WHEEL-LOADER CAB SIMULATION

The earth-moving machine is a W130 Fiat-Hitachi. The original cab FE model (Figure 1) consisted of 11461 shell elements. This model has been simplified mainly for two reasons:

- the computation time should have been too long [(1)];
- many insignificant details were present from the acoustic point of view.

A surface reconstruction was needed to include into the FE model the geometrical entities too. This implementation has to be considered carefully. Much time could be spent for this necessary operation. The reconstruction has been carried out partly by direct commands and partly with the pre-processing tools developed in Ansys.

Steel and glass have been identified as the two most important materials. Their properties are described in Table 1.

Material	E (MPa)	ν	ρ (kg/m^3)
Glass	65000	0.23	2500
Steel	206800	0.29	7830

Table 1: Materials characteristics

The stiffening tubes, present in the real cab, have been represented with linear beam elements, their properties being derived from the characteristics of the original model. Their effect on the structural behaviour must be taken into account, owing to a natural frequency shift of about 2 Hz for the complete model with and without tubes. The beam elements are directly connected to the shell ones as the corresponding nodes are identical for both.

A mesh of 2462 elements (Figure 2) has been created and used in the numerical computation optimisation phase. Discretization errors have been considered comparing the results (sound pressure levels, or SPLs, at the operator ear position) obtained with two different meshes (2462 elements and about 5000 elements). The acoustic accuracy has been also assured by considering the ‘at least 6 elements per wavelength’ rule of thumb. It turned out that the explored spectrum highest frequencies could not be precisely evaluated using the 2462 elements mesh. Being the results quite close and satisfactory especially as far as the lowest frequencies of interest are

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[(1)] the computation times for the vibro-acoustic simulation grows approximately as n^2 or n^3, being n the number of elements. It has to be considered that the computation of 21 frequencies - the ones required for the optimisation process - took about 40 min.

concerned, the 2462 elements mesh has been chosen to be used for the further computations.

In Table 2 and Figure 3 the structural results are represented calculated with a mesh of about 5000 elements (the agreement with the coarser mesh has been verified).

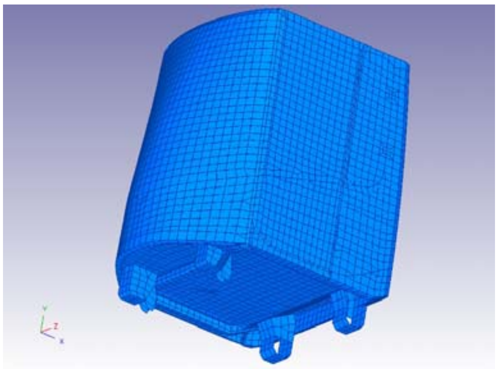


Figure 1: Cab 3D original design FE model

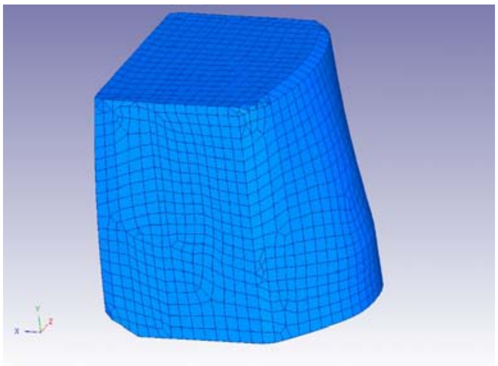
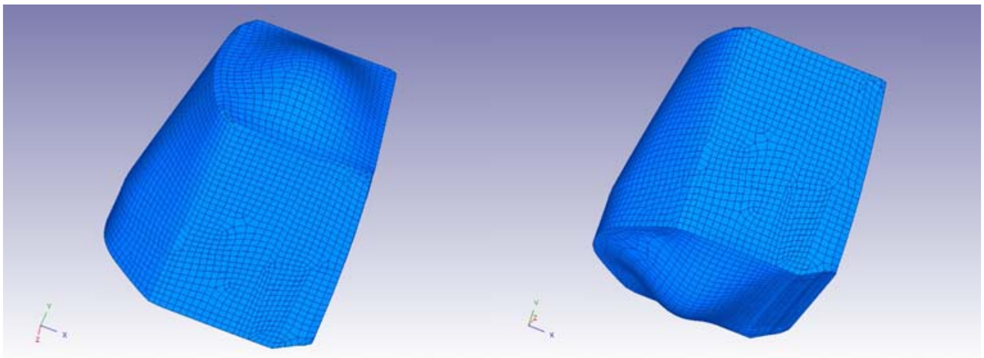


Figure 2: Analysis and optimisation cab FE model

1	**2**	**3**	**4**	**5**	**6**	**7**
12.040	16.011	17.120	20.693	23.120	25.544	25.925
8	**9**	**10**	**11**	**12**	**13**	**14**
26.866	27.249	31.542	34.044	34.962	36.078	36.642

Table 2: First 14 calculated natural frequencies (Hz)



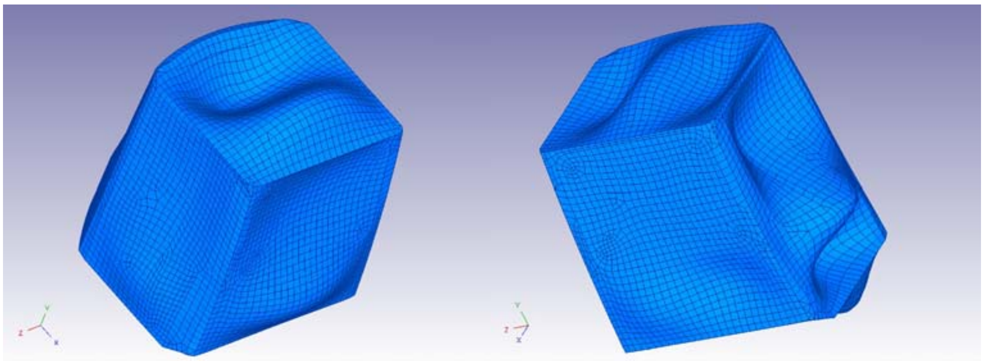


Figure 3: Structural mode shapes. a) 12.0 Hz, b) 16.0 Hz, c) 25.9 Hz, d) 39.1 Hz

2. EXPERIMENTAL MEASUREMENTS

The experimental measurements (Figure 4) are useful in this context as a reference both for the acceleration values - needed as inputs to force identification and therefore to numerical analysis - and for the SPL values - considered the targets to be matched by computations. Moreover, they are helpful in order to explore the various vibro-acoustic factors and verify the related hypotheses.

The acceleration values at 16 points located on the cab base and panels and the sound pressure levels at the operator ear position have been acquired between 0-3 kHz in third-octave bands. A Larson & Davis 2900B 2-channel data acquisition system has been used. The engine was maintained at slow running speed. From the diagrams (Figures 5 and 6), it turned out that the peak values both for accelerations and SPLs appear at 40 Hz, 160 Hz and 315 Hz, suggesting that the fluid-structure coupling is significant. As far as the accelerations are concerned, a peak appears also at 800 Hz. These frequencies correspond to the engine harmonic frequencies. In particular, 40 Hz is the value of the first harmonic frequency for a 4-strokes engine running at 1200 rpm.

Another interesting result came out from the comparison between the SPLs measured with the lateral door panels closed and opened, respectively. A difference approximately constant and not very remarkable turned out to exist between these two spectra. It can not be stated surely that this difference equals to the air-borne noise contribution, that is present also with the cab panels closed. Anyway, this trend suggests that the measured noise field is mainly due to the structure-borne noise contribution.

Figure 4: Wheel-loader cab and measurement set-up

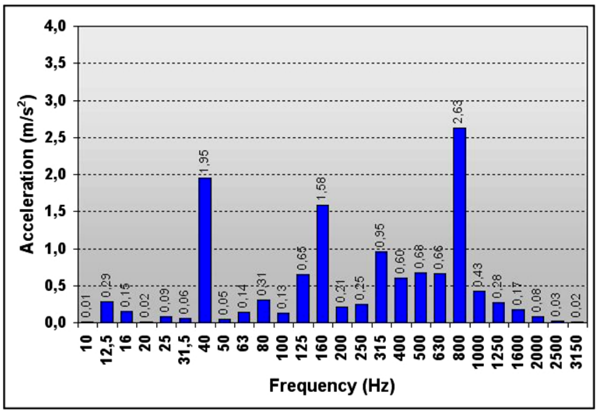


Figure 5: Cab base point acceleration level spectrum

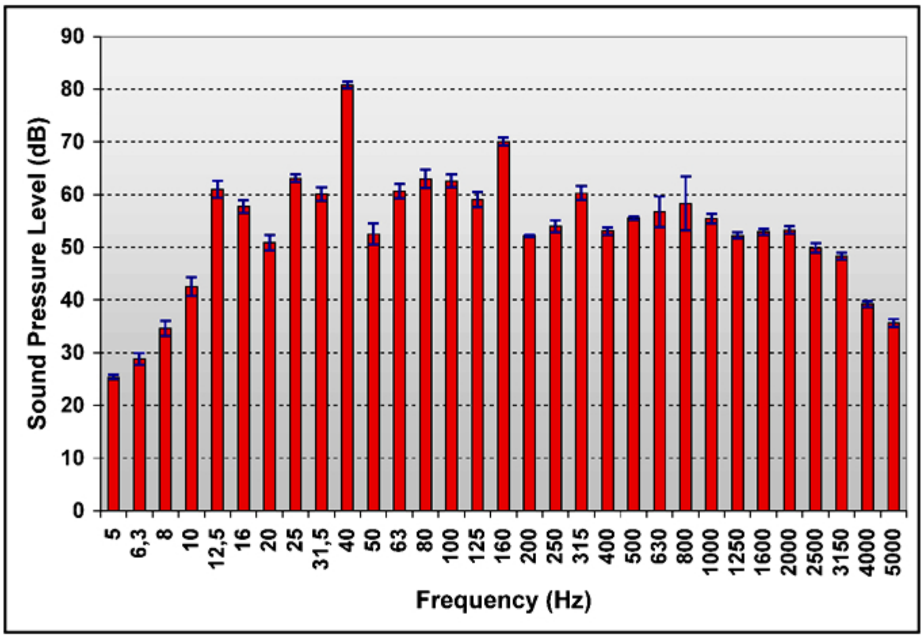


Figure 6: Operator ear position SPL spectrum and standard deviation

3. ANALYSIS AND COMPUTATIONAL RESULTS

1. Fluid-structure coupling

The interpretation of physical phenomena is always very important when a numerical analysis is carried out. As far as the vibro-acoustic problems are concerned, the degree of fluid-structure interaction must be considered. In the uncoupled problems a one-way coupling is considered: for a vehicle cab, this means not taking into account the fluid loads acting on the structure, but only the loads generated by the structural vibrations. For this application, a strong coupling fluid-structure interaction has been considered in the numerical computations, based on the following criteria:

- close structural and acoustic mode frequencies;
- similar structural and acoustic mode shapes, considering a quite regular volume like the one of the cab;
- value of the dimensionless number λ_c[2]: if $\lambda_c>1$ a strong coupling can be expected, if $\lambda_c<<1$ the uncoupled model can be a sufficient approximation.

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[2] λ_c is defined as the ratio between the quantities describing the fluid behaviour (ρ_0 and c: density and speed of sound), multiplied each other, and the quantities describing the structural

For this application λ_c=0.2: rather difficult to decide if coupling has to be considered or not. λ_c does not include the cab dimensions; since these are quite small, it turns out to be a well-founded reason to consider the load effect of the air pressure acting on the panels.

From a theoretical point of view, this choice consists in taking into account the wholly coupled equations as reported in [6].

2. Numerical analysis

A Sysnoise indirect BEM approach has been chosen for the vibro-acoustic computation. This procedure turns out to be faster than the direct BEM one as far as large problems are concerned, owing to the presence of symmetric matrices. A FEM computation procedure has been applied to the structural model. The two models have been linked together in order to solve the coupled problem. The solution has been computed referring to modal coordinates, the structural model modal basis imported from Ansys.

The beam elements can not be used for the acoustic simulation since the shell element normals are not consistent with a Sysnoise BEM interior analysis. For this reason, the modal uncoupled basis has been computed in Ansys taking into account the complete mesh. On the other hand, the vibro-acoustic field has been evaluated in Sysnoise with no beam present. This approximation turns out to be acceptable, the main effect of the beams consisting indeed in a structural modification and not in a different acoustic radiation.

3. Force identification

The forces acting on the mounts on the base panel of the cab has been identified by a dedicated procedure in FEMtools [7]. This analytical computation has been developed for two reasons:

- with the actual configuration of the cab and the available instruments it was impossible to measure directly the values of the excitations in a direction normal to the cab base;
- a computing procedure independent from the variation of the parameters (geometry and thicknesses) was needed for the optimisation; this was not the case of an analysis starting from the panel accelerations and defining velocity boundary conditions on the cab surfaces, while the structural

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behaviour (ρ_s, t and ω: density, characteristic thickness and circular frequency of the excitation), multiplied each other.

excitations can be considered independent from the variation of the cab design with good approximation.

The force identification was possible taking into account the measured accelerations and the FE model. The accelerations at seven points of the cab base were experimentally measured, being strictly connected with the excitations at the mounts. The measurement points were linked to the corresponding nodes of the FE model and the forces were computed by FEMtools. This is possible by using the appropriate equations that link the forces values with a limited number of structural responses by means of the calculated mode shapes. More detailed information about the procedures concerned and the theoretical background can be find in [7] and [8]. The resulting complex force values have been used as boundary conditions to the structural part of the model in the vibro-acoustic computations.

4. Model validation

The model has been validated by comparing the sound pressure level measured and calculated values at the operator ear position; the results of this procedure are represented in Figure 7. In this context, describing carefully the effects of a parameter variation on sound pressure levels is more important than achieving a very accurate validation of the parameter numerical values that match the experimental target.

Since the geometrical parameters and the materials properties have been described as precisely as possible from the actual knowledge of the model, the effect of the admittance boundary conditions associated to the presence of the acoustic absorbing panels (located on the real cab roof and front panel) has been investigated. The admittance values have been calculated as complex numbers depending on the acoustic absorption coefficient, varying with frequency [9] and whose trend is represented in Figure 8.

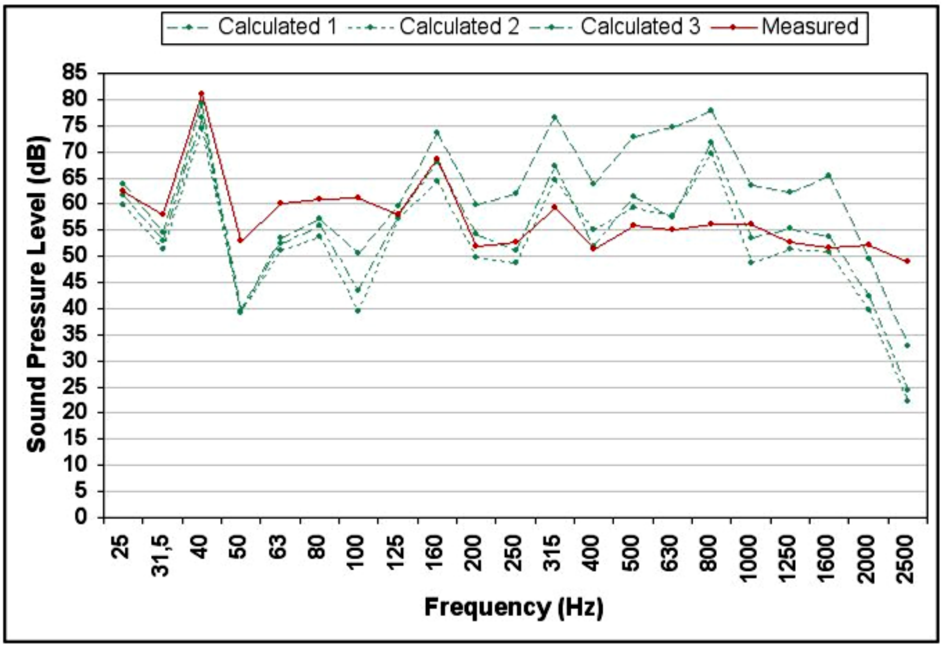

Figure 7: Operator ear position SPL comparison. Computed (green), measured (red)

At first, these admittance boundary conditions have been applied to the surface corresponding to the real absorbing panels (‘calculated 1’ in Figure 7). In this case the obtained sound pressure levels resulted too high especially for the highest frequencies. These results were compared with those calculated by applying the admittance boundary conditions on the whole cab (idealized situation represented as ‘calculated 2’ in Figure 7), obtaining a better agreement with the measured results. From this exercise, it was observed that the absorbing conditions must be accurately considered for the calculation of the interior acoustic field and it was observed that a global effect of acoustic absorption is missing by considering only the admittance values related with the real absorbing surfaces. This effect is due to the peculiarities of the model, in which the absorbing properties of the interior elements (i.e. seat, plastic panels and others) are not considered.

For these reasons, the best results have been obtained supposing that a minor, but different from zero, absorption coefficient has to be defined also for the smooth surfaces. This trend can be seen in the spectrum indicated as ‘calculated 3’ in Figure 7. These results have been obtained considering different values of the absorption coefficient. On the surfaces where the sound absorbing material is really present, the absorption coefficient has been set as referred in Figure 8; for the remaining area it has been assumed to be equal to half of the previous one.

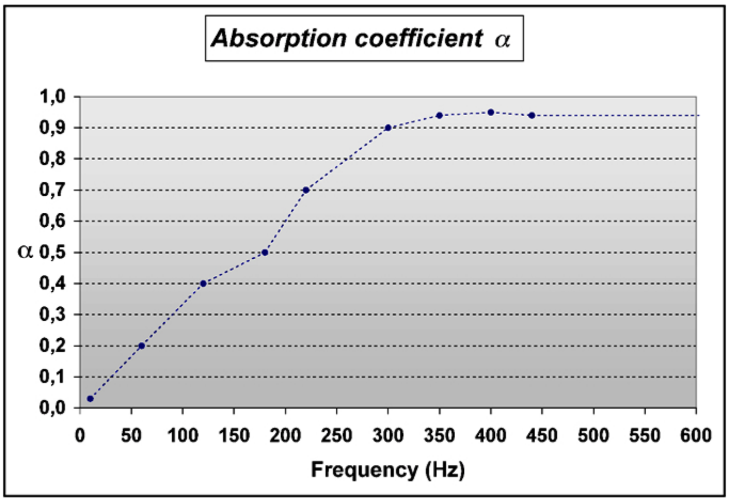

Figure 8: Soundproofing material absorption coefficient

4. VIBRO-ACOUSTIC OPTIMISATION

The numerical optimisation has been realized with modeFrontier by means of a genetic algorithm (MOGA-steady evolution). The DOE representing the first generation of individuals was defined with a Sobol methodology. The general objective was the minimization of the spectrum amplitude representing the sound pressure level at the operator ear position. This optimisation goal was splitted into four objectives to minimize: three frequency bands and the peak value at 40 Hz. For each band the sum of the sound pressure levels was considered. Six constraints were also defined. Four of these were related to the values of the objective variables and the two remaining referred to the peak values at 160 Hz and 315 Hz.

At the beginning five parameters were considered:

- the thickness of the steel panels;
- the thickness of the glass panels;
- the thickness of the stiffening tubes;
- the global dimensions (introducing a scale factor);
- the location of an extra absorbing panel.

The role of the last parameter is easily understood, considering the global effect of adding sound proofing material on different parts of the cab. Five possible zones have been defined, on the lateral doors, on the lateral panels, and on the back panel.

From statistic evaluations (Student parameter) the thickness of the tubes resulted to be less important than the other variables. The other thickness parameters tended to reach the highest values in the Pareto designs, while the trend of the parameter related to the absorbing panels was not clearly defined, even if this parameter resulted to be effectively significative.

Moreover, maximizing the thicknesses results in a consistent cab mass growth, that is an “unfeasible” design for the analyst. For these reasons, another run has been carried out in order to investigate more precisely the most important parameters; an additional constraint limiting the total cab mass has been set. Three parameters were considered, related to the same objectives:

- the steel thickness, on the lateral panels;
- the steel thickness, on the roof, on the back panel, on the base and on the front panel;
- the location of an additive absorbing panel.

In the following figures the reductions are represented of the predicted SPL at the operator ear location obtained by the optimisation. Figures 9 and 10 show the spectra of the 20 Pareto frontier designs computed in the first run: the reduction is consistent in every part of the spectrum. Figure 11 shows the spectra of the 6 Pareto frontier designs computed in the second run: the reduction is not so evident but it is still consistent in every part of the spectrum. Moreover, it is satisfying that the peak value at 40 Hz can be reduced of about 5-6 dB, by reducing at the same time also the other spectrum pressure levels. This result is obtained mainly acting on the most significative parameters. These turned out to be the steel thickness and the admittance values, among the ones considered. Table 3 shows some statistics about computational runs.

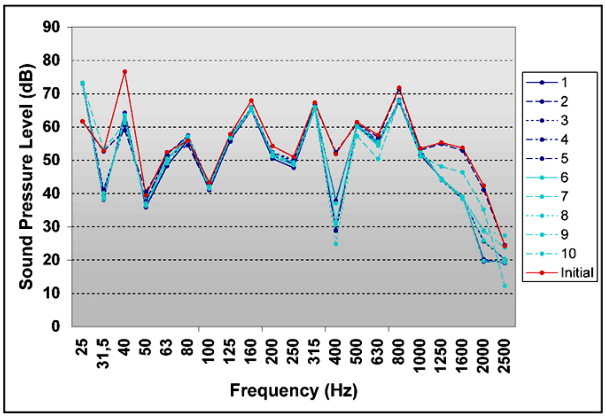

Figure 9: Operator ear position SPL comparison. Pareto frontier designs (1 to 10)

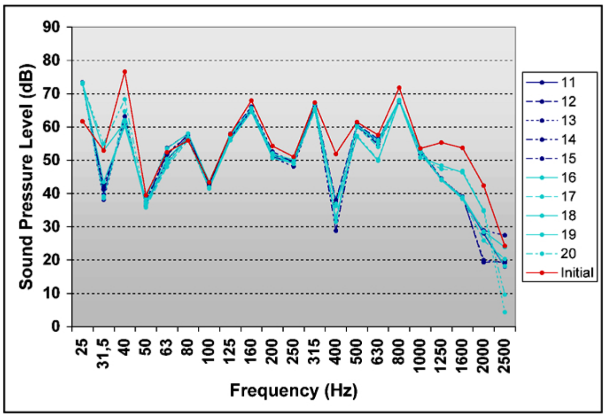

Figure 10: **Operator ear position SPL comparison. Pareto frontier designs (11 to 20)**

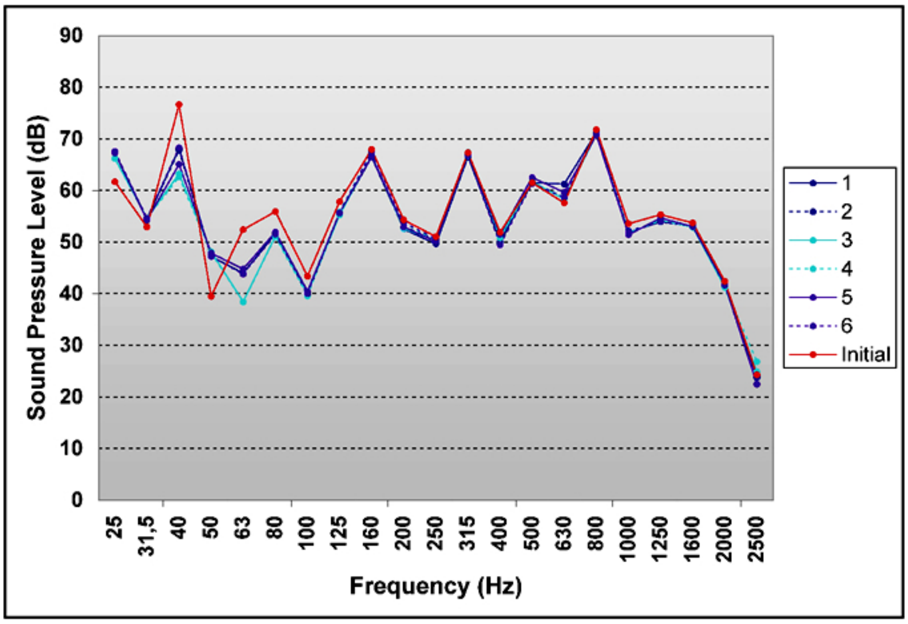

Figure 11: **Operator ear position SPL comparison. Pareto frontier des. (1 to 6, refining)**

Computational run n.	1	2 (refining)
Number of DOE elements	36	24
Number of individuals’ generations	9	7
Total number of designs evaluated	324	168
Computation time for 1 design (21 frequencies)	39’30”	39’30”
Total computation time	90 hours	40 hours
Not repeated designs	122	60
Pareto designs	20	6

Table 3: Computation statistics (workstation ORIGIN 200)

5. CONCLUSIONS

A general procedure for a construction machinery cab vibro-acoustic numerical analysis and optimisation has been set up and verified. Different software tools have been combined in an integrated optimisation tool capable of supporting experimental evidence and multiple constrained optimisation runs.

The procedure proved to be reliable and quite efficient and could be applied in order to improve the vibro-acoustic behaviour of existing products or in the early design phase of other similar equipments.

Future work will be mostly focused on the improvement of the mechanical modelling of the structure and the force definition together with the impedance characterization of the sound absorbing materials.

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