ABSTRACT

The present work deals with the numerical prediction of the combustion noise of a newly developing aeronautical diesel engine. The study has been performed on a three dimensional basis by the integration of FEM and BEM codes integration, in order to correlate the radiated noise with the combustion pressure distribution inside the cylinders. The last purpose of such a procedure is the characterization of the engine in terms of acoustic power to be determine according to normative ISO 3746.

The set of actions that have been performed may be summarized as follows:
1. Starting from the CAD model, a 3-D finite element model of the engine has been realized;
2. The model has been characterized in terms of eigen-frequencies and modes
3. The surface vibration velocity has been measured under the in-cylinder combustion pressure load (at the investigated engine speeds);
4. Direct BEM approach has been than that used to evaluate the radiated noise field in terms of sound pressure at 1 meter of distance and in terms of sound power according to the ISO 3746.

These results have been used as driving parameters for successive engine structural design and will be later on experimentally verified.

FEM/BEM Approach

The test article presented in the paper is represented by a newly developing aeronautical diesel engine with 6 cylinders disposed to V with an angle equal to 180°. The engine is symmetric in respect to the axes y, as shown in figure 1. In this way, it has been possible to realize a FE Model of half engine. This condition has the advantage to study half propeller, aiming to reduce calculation time. The geometrical model has been, therefore, modified by the Solidworks 2004 software for the preparation of this mesh process; in particular, the calculation model will be constituted by the half engine, as it is visible in figure 1.
The structure of the present activity is shown in the diagram 1.

Figure 2. - Activity scheme

The mesh process has been realized subdividing the model in single group, in order to work with single part, and subsequently assembled. In particular it has been divided in three main parts: top of cylinders, cartridges and engine. In figure 3 is reported the two first mesh models. In particular, it can be noted the complexity to mesh the cartridges because of the presence of numerous holes and stiffening surface.

Figure 3. – Mesh models

The final step has been the realization of the mesh for the greater part of the engine, represented in figure 4.

Figure 4. –Mesh model of the engine

The three different models have been then assembled and the final mesh model is represented in figure 5.
The model consists therefore in a total of **921759** elements and **782743** grid. The model, so discretized, has been object of different types of analyses; in particular a real modal analysis has been effected up to the frequency of 3500 Hz, then, a frequency response analysis has subsequently, been conducted using as excitation the pressures that acting inside the cylinders during the combustion process at 2500 rpm engine speed (critical engine rotation). Such data have been gotten by mono-dimensional computational analysis.

The select code for such types of analyses is represented by the commercial code MSC/Nastran®. In table 1, are reported the first 10 natural frequencies.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>441</td>
</tr>
<tr>
<td>2</td>
<td>531</td>
</tr>
<tr>
<td>3</td>
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</tr>
<tr>
<td>8</td>
<td>1360</td>
</tr>
<tr>
<td>9</td>
<td>1412</td>
</tr>
<tr>
<td>10</td>
<td>1526</td>
</tr>
</tbody>
</table>

Table 1. – Natural frequencies

In figure 6 are shown the first two deformation carried out by the modal analysis. The computational time for such analysis has been of about 10 hours because of the great number of nodes used for such mesh model.

With the purpose to get information on the superficial vibrations of the skin (external skin) of the model, the frequency response analysis has been effected defining a distribution of the loads most possible realistic. To such intention, the cylinder pressure cycles have been used which are obviously tied up to the alternative pistons motion. Such motion imposes therefore the different loads acting in the cylinder grids during the time \( t \), more precisely the subdivision of the grids above the piston solicited by the pressure present during the cycle engine and of those below solicited instead by a reference pressure value in the carter that can be set equal to the atmospheric pressure. Since the grids to be loaded in this way would be always different varying time \( t \) of the cycle it is useful to refer of an equivalent pressure \( p_{eq} \) variable in the time but uniform in the space to apply to the whole internal surface of the single cylinder.
As shown by the sketch in the figure 7, it is able, therefore, to divide the internal cylinder surface in two parts: the heading of the cylinder that will be solicited by the real instant value of pressure in combustion chamber (represented by the red line in figure) and the side surface (green line) that it will be interested from the equivalent pressure calculated according to the following relationship:

Figure 7. - Schematic sketch of the cylinder for the pressures calculation

\[
p_{eq} V_{tot} = p_{cyl} V_{cyl} + p_{ref} V_{ref}
\]

\[
p_{eq} \pi D (h_0 + s) = p_{cyl} \pi D h_1 + p_{ref} \pi D h_2 \quad (\text{Eq. 1})
\]

\[
p_{eq} = \left( p_{cyl} - p_{ref} \right) \frac{h_1}{h_0 + s} + p_{ref}
\]

where \(D\) represents the bore, \(V_{tot}\), \(V_{ref}\) and \(V_{cyl}\) respectively represent the total volume of the cylinder and the suitable volumes in the previous figure, \(h_0\), \(h_1\) and \(h_2\) represent the heights of the combustion chamber and the portions of cylinder of figure and \(s\) the piston run. In figure 8, are reported the total and equivalent pressures drawn by the equation 1.

Figure 8. Effective and equivalent pressures (Pa) vs crank angle acting in the three cylinders

In figure 9 is represented the spectral analysis of the cylinder pressure utilized as excitation in the frequency response analysis to obtain information on the surface velocity on the skin surface.

Figure 9. – Spectral analysis, amplitude (right) and phase (left) of the cylinder pressure.
The frequency analysis response has been conducted in the frequency range 40-2000 Hz step 20 Hz to evaluate the superficial vibrations of the engine that act during the combustion process. In parallel to the calculation of surface velocity (calculation time of about 11 hours), it has been realized a BEM model with a reduced number of nodes and elements. The data of vibrational output so obtained will represent the boundary conditions to be applied to the BEM model for the final radiated sound power calculation. It is constituted by 9483 grid and 11912 elements, of which 4403 type TRIA3 and 5509 type QUAD4. The software used for the BEM methodology is the commercial computational code LMS/Sysnoise. The approach used in the present work is represented by the technology ATV (Acoustic Transfer Vectors). Such technology, through the preliminary evaluation of the functions of transfer surface-receiver (microphone), allows to evaluate in real time the answer to different boundary conditions: application fine-load case and multi-frequency (engine noise); inverse acoustics (structural vibration determination of the pressure measurements); contribution of panels etc. Synthetically the relationship can be written:

\[
\{\text{Sound Pressure}\} = \left[\text{AcousticTransferMatrix}\right] \cdot \left\{\text{SurfaceVelocities}\right\} \quad (\text{Eq. 2})
\]

where \{\text{Sound Pressure}\} is a vectorial column containing the acoustic pressures in different computational points for the external acoustic radiation, \{\text{SurfaceVelocities}\} is a vectorial column containing the structural velocities of the vibrating panels and \left[\text{AcousticTransferMatrix}\right] is the matrix system of input and output data. More precisely, surface velocities are represented by the normal components to the elements of the structural velocities coming from the FEM calculation of frequency response analysis and they play an important role for the generation of the noise waves produced by them. Representing, therefore, with \{v_{ns}\} the structural velocities and considering the sound pressure level \(p\) in a single point of measurement, the following relationship dependent tightly by the frequency \(\omega\) can be written:

\[
\{v_{ns}\} = \langle\text{ATV}\rangle \cdot \{v_{ns}\} \quad (\text{Eq. 3})
\]

where ATV represents the abbreviation of Acoustic Transfer Vector (vectors of acoustic contribution) that represent acoustic transfer function tied up to the normal vibration velocities on a discreet number of panels and to the pressure of acoustic radiation in a single microphone, as illustrated in the following figure.

\[
L_W = L_p + 10 \log \left(\frac{S}{S_0}\right) \quad (\text{Eq. 4})
\]

where: \(L_W\) = acoustic power level (dB); \(S\) = measurement surface (rif. \(S_0=1\ \text{m}^2\)); \(L_p\) = sound pressure level emitted by the radiant surface express as:
\[ L_p = 10 \log \left( \frac{1}{N} \sum 10^{0.1L_{pi}} \right) \]  
(Eq. 5)

N=number of microphone positions; Lpi=sound pressure level in the i-mo microphone. In base of such formulation, and as previously described, a measurement surface of the power is created around the engine model according to normative ISO (figure 12).

**CONCLUSION**

Within the present work that has like objective the estimation of the noise produced by the engine under investigation, and downstream of the fluid-dynamic results and to the vibro-acoustic analysis developed through the integration of FEM-BEM codes, has been estimated an acoustic pressure level according to normative ISO 3744, to 1 meter of distance from it, equal about 95 dB. Future experimental investigations are planned in order to eventually update the numerical model.

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**References:**