

Noise emission prediction from Co-Generator. Comparison between SEA modeling and OTL Propagation model based on spherical waves

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Abstract

The present work is based on a previous publication (see in Ref.14) covering the whole process to predict the noise pollution impact of a new Co-Generator located in a mixed industrial and residential area in a residential area.

Here we focus more to discuss the comparison between two forecasting methods of calculation: SEA and OTL_SWP (SWP as Spherical Waves Propagation). Both simulations consider the container box divided in sub-systems characterized by sound energy generator, boundaries impedance, coupling factors among them and with the outside.

1 Introduction

Upon acoustic data available from various suppliers (engine, exhaust, mufflers, fans, pumps, etc.) it has been estimated the Sound Power of each source in order to apply SEA method (Statistical Energy Analysis) to check for propagation and paths.

Exhaust tail (chimney) source is defined as a point source without directivity, exhaust piping is estimated as linear source; sound emission from cooling tower and energy absorber has not been taken into account because of their low acoustic impact.

Let us recall the two main calculations:

- a propagation model from Sound Pressure Level LpSEA obtained using semi-infinite fluid

propagation and Lpmodel at selected receptors for the residential area;

- a propagation model based on spherical waves propagation within the industrial plant area up to selected receptors in the residential area.

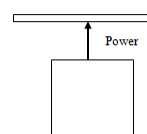
Principal Noise sources have been recognized and classified with an experimental approach based on the estimated Sound Power Level - or Lp depending from different technical documents - and the emission area extension.

Category 1 has the higher contribution (typically engine box and chimney) followed by muffler and exhaust line.

2 A simplified introduction to SEA ⁽¹⁾

Statistical Energy Analysis and Power Balance, a simplest case:

- a shaker injecting some force on a steel plate without connections to the neighbouring structures.



The supplied power must be equal to the sum of power lost through plate internal damping and power radiated as sound in the surrounding medium (air).

Total energy E stored in the plate equals the sum of kinetic and potential energies for the plate (equal in average).

$$E = 2 \cdot \left(0.5 M \langle v^2 \rangle \right)$$

M = plate mass kg;
 v^2 = mean squared trasvers vibrational velocity averaged across the plate

We can also write the equation in a different form:

Energy dissipated in the plate in one cycle of vibration is equal to the total energy stored E times the dissipation loss factor.

$$E = \rho_s S \langle v^2 \rangle$$

ρ_s = plate mass per unit area kg/m²;
 S = plate area in m²

$$\omega W^d = E \omega \eta_d$$

ω = angular frequency;
 η_d = dissipation loss factor

Acoustic power radiated by a 2 side plate in free space, is equal to the mean square velocity *times* the characteristic resistance ρc_0 of the air *times* the radiation efficiency σ_{rad} .

$$W^{rad} = \langle v^2 \rangle 2S \rho c_0 \sigma_{rad} \quad \text{watt}$$

If the plate has rigid connections (structure born) energy is lost at the boundaries, this loss can be taken in account by increasing the dissipation loss factor to $\eta_t > \eta_d$

If we divide the radiated power by the injected one, we obtain the power transfer from the source to the air.

$$\frac{W^{rad}}{W_{in}} = \frac{2\rho c_{0rad}}{(\rho_s \omega \eta_t + 2\rho c_{0rad})}$$

If losses in the plate are very large most of the power is dissipated in the plate

$$W^{rad} = W_{in} \frac{2\rho c_{0rad}}{\rho_s \omega \eta_t}$$

If losses in the plate are very small; $W^{rad} \cong W_{in}$

SEA method allows to correctly calculate energy flows between connected resonant structures as well as between plates and reverberant sound field in an enclosure.

Giving the energy E stored in a structure, in SEA we define *system* the ensemble of resonant modes (or resonator). If we assume the power flow between two groups of resonant modes of two coupled structure having resonant frequencies in a band Δ_ω , we can write the balance equation.

$$W_{12} = W'_{12} - W'_{21} = E_1 \omega \eta_{12} - E_2 \omega \eta_{21}$$

Assuming that each resonant mode of system 1 has the same energy coupling of individual resonant modes of first system and the ones of system 2 are approximately the same waves carrying energy in one system, are uncorrelated with the ones carrying energy gained through coupling in the other system, than we can separate the power flow.

The net power flow between the two system is described by Balance Equation:

$$W_{12} = W'_{12} - W'_{21} = E_1 \omega \eta_{12} - E_2 \omega \eta_{21}$$

and we define Modal Energy as:

$$E_m = \frac{E(\Delta_\omega)}{n(\omega)\Delta_\omega} \quad \text{watt - sec/ Hz}$$

Upon all valid previous assumptions we may write:

$$\frac{\eta_{21}}{\eta_{12}} = \frac{\eta_1(\omega)}{\eta_2(\omega)}$$

which implies that for equal energy in the two systems, the system with lower modal density transfer more energy to the other one than vice-versa.

Terms definitions:

W'_{12} = powers system 1 transmitt ed to system 2, watts

W'_{21} = powers system 2 transmitt ed to system 1, watts

E_1 = total energy in system 1, m - Kg/sec

E_2 = total energy in system 2, m - Kg/sec

η_{12} = coupling loss factor from system 1 to system 2

η_{21} = coupling loss factor from system 2 to system 1

$E(\Delta_\omega)$ = total energy in the system in the angular frequency band Δ_ω

$n(\omega)$ = modal density = number of modes in unit bandwidth

$(\Delta_\omega = 1)$ centered in ω

Δ_ω = bandwidth, radians/se c

$\eta_1(\omega)$ = modaldensityof system1 at frequency ω ,sec

$\eta_2(\omega)$ = modaldensityof system2 at frequency ω ,sec

Combining equations:

$$\begin{cases} W_{12} = W'_{12} - W'_{21} = E_1 \omega \eta_{12} - E_2 \omega \eta_{21} \\ \frac{\eta_{21}}{\eta_{12}} = \frac{\eta_1(\omega)}{\eta_2(\omega)} \end{cases}$$

we obtain:

$$W'_{12} = E_1 \omega \eta_{12}$$

$$W'_{21} = E_2 \omega \eta_{21}$$

$$W_{12} = \omega \eta_{12} \eta_1(\omega) [E_{m_1} - E_{m_2}] \Delta \omega$$

The equation is positive if the first term in the brackets is greater than the second.

The SEA method raw principle consists in a simple algebraic equation with energy as independent variable.

In a narrow frequency band centred in ω the net power flow between two coupled systems is proportional to the difference in modal energies of the two systems at the same frequency. Energy flows from the system with higher modal energy to the one with lower modal energy.

The case of equal modal energies is equivalent to bodies having the same temperature. Considering thermal-acoustic analogy, heat flows between two bodies with different temperatures, from the one having higher temperature to the one having lower temperature and the heat net flow is proportional to the temperature difference.

$\Rightarrow$ Modal energy E_m is the temperature.

$\Rightarrow$ Net power flow W_{12} is the heat flow.

3 Fundamentals conceptual notes

Equal energy of vibration modes: they will exist within a group if the wave field in the structure is diffuse; since adjacent modes are coupled by scattering and damping, there is always a tendency for modes modal energy to equalize, even in a non diffuse field.

Non correlation between waves usually, in sound transmission problems only one system is

usually excited; power transmitted W_{12} to the non-excited system builds up a semi-diffuse vibration fields in that system. Waves carrying back transmitted power $-W_{21}$ to the excited system are sufficiently delayed and randomize in phase to the first one, so there is a little correlation between the two.

Equal coupling loss factor between individual modes within a group is a matter of similar nature modes; if the coupled system is a plate in a reverberant sound field, the existence of phenomena like “acoustically slow edge and corner modes” as well as “acoustically fast surface modes” grant the concept of grouping.

We will show that radiation ratio is closely connected to the coupling loss factor.

Composite structures made up of plates, beams, stiffeners, etc. can be divided into simple members whenever the wavelength of the structure-borne vibration is small compared.

In this case the complex structure modal density approximately equals the elements modal densities sum. Providing to know the power input as well as various coupling loss factors and the power dissipated in each element, the power balance equation yields the vibration energy of the structure respective elements.

Dissipative loss factors of each element are obtained by separating each element from the rest of the structure and measuring its decay rate.

in a two-structure system the *Power balance* is given by two algebraic equations:

$$W_1^{in} = W_1^d + W_{12}$$

$$W_2^{in} + W_{12} = W_2^d$$

Terms definitions:

W_1^{in} = input power to system 1

W_1^d = power dissipated in system 1

$W_{12} = W'_{12} - W'_{21}$ = net power lost by system 1 through coupling to system 2

W_2^{in} = input power to system 2

W_2^d = power dissipated in system 2

Power dissipated in a system is related to the energy stored by that system E_i through the dissipative loss factor η_i :

$$W_i^d = E_i \omega \eta_i$$

where: E_i = energy stored in system i

Assuming that the second system does not have any power input ($W_2^{in}=0$) and combining preceding equations ($i=2$) we arrive to the ratio of energies stored in the two systems:

$$\frac{E_2}{E_1} = \frac{\eta_2}{\eta_1} \frac{\eta_{12}}{\eta_{21} + \eta_2}$$

If the coupling loss factor is very large compared to system 2, then: $\eta_{21} \gg \eta_2$. The equation yields the equality of the modal energies.

Diffuse Sound Field driving a freely hung panel
Let's now consider again two systems: System (1) is a reverberant room in which persist a *Diffuse Sound Field driving a freely hung panel* System (2).

Total energy for each system is than given by:

$$E_1 = DV = \frac{\langle p^2 \rangle}{\rho c^2} V$$

$$E_2 = \langle v^2 \rangle \rho_s S$$

D = average energy density in the reverberant room;
 $D = \frac{4W}{cR}$; $R = \frac{S\alpha}{1-\alpha}$
 $\langle p^2 \rangle$ = mean square sound pressure (space-time average), $\frac{N^2}{m^4}$
 V = room volume, m^3
 $\langle v^2 \rangle$ mean square plate vibration velocity

In order to find the coupling loss factor η_{21} we note that W_{21} equals the power that the vibrating plate radiates back in the room. Thus $W_{21} = W_{rad}$.

Combining the following two equations:

$$\begin{cases} W_{rad} = \rho c S \sigma_{rad} \langle v^2(\omega) \rangle \\ W_i^d = E_i \omega \eta_i ; i = 2 \end{cases}$$

with Step 1, we obtain:

$$W_{rad} = W_{21} = 2 \langle v^2 \rangle \rho c \sigma_{rad} S \equiv E_2 \omega \eta_{21} = \langle v^2 \rangle \rho_s S \omega \eta_{21}$$

$$\text{Solving for } \eta_{21}: \quad \eta_{21} = \frac{2 \rho c \sigma_{rad}}{\rho_s \omega}$$

σ_{rad} = radiation ratio for the plate
 W_{rad} = acoustical power radiated from both side of the plate

The modal densities of the reverberant sound field in the room $\eta_1(\omega)$ and of the thin homogeneous plate $\eta_2(\omega)$ are:

$$n_1(\omega) = \frac{\omega^2 V}{2\pi^2 c^3} \quad c_L = \text{propagation speed of longitudinal waves in the plate material, m/s}$$

$$n_2(\omega) = \frac{\sqrt{12} S}{4\pi c_L t} \quad t = \text{plate thickness, m}$$

Inserting from:

$$E_1 = DV = \frac{\langle p^2 \rangle}{\rho c^2} V$$

$$E_2 = \langle v^2 \rangle \rho_s S$$

We get the relation between Sound Pressure and Plate velocity

$$\langle v^2 \rangle = \langle p^2 \rangle \frac{\sqrt{12} \pi c^2}{2 \rho c t c_L \rho_s \omega^2} \frac{1}{1 + \rho_s \omega \eta_2 / 2 \rho c \sigma_{rad}}$$

The panel mean square acceleration (space-time average) of the panel is:

$$\langle a^2 \rangle = \omega^2 \langle v^2 \rangle = \langle p^2 \rangle \frac{\sqrt{12} \pi c^2}{2 \rho c t c_L \rho_s} \frac{1}{1 + \rho_s \omega \eta_2 / 2 \rho c \sigma_{rad}}$$

As long as power dissipated in the plate is small compared with sound power radiated by the plate we obtain:

$$\rho_s \omega \eta_2 \ll 2 \sigma c \sigma_{rad}$$

The sound field and plate modal en-ergies equality yields the proper plate velocity and acceleration; in this con-dition the ratio between the mean square plate acceleration and the mean square sound pressure is inde-pendent from the frequency.

It is important to underline that the plate response is always smaller than the one calculated by the equality of the modal energies by the factor:

$$\frac{1}{1 + \rho_s \omega \eta_2 / 2 \rho c \sigma_{rad}}$$

which equals the ratio of the power loss by acoustical radiation to the total power loss.

4 The OTL- model geometric propagation

Olive Tree Lab (OTL) – Terrain [4] is a geometrical acoustics wave based software which simulates sound propagation using advanced calculation methods. It utilizes sound propagation modeling solving Helmholtz's sound wave equation and it accounts for recursive sound diffraction / reflection of any order. It accounts for reflection from finite size surfaces of finite impedance using Fresnel zones and spherical wave reflection coefficient concepts respectively. The latter allows the Ground Wave analysis.

It takes also into account coherent and incoherent source contribution, distance attenuation, atmospheric absorption, and atmospheric turbulence.

These embedded features allow in deep wave interference phenomena study in resolutions down to 0.001Hz within a frequency range 0.001 Hz – 100kHz. It also provides Impulse Response and Auralisation.

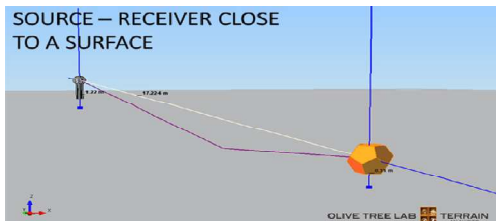


FIGURE1: SOURCE-RECEIVER CLOSE TO AN INFINITE SURFACE OF FINITE IMPEDANCE, RC 200 KPAS M⁻²

OTL-Terrain is based on Salomons [5] work, applying a propagation model using analytical solutions. Spherical wave diffraction coefficients are given by Hadden and Pierce [6]. Spherical wave reflection coefficients are based on the work of Embleton [7], while ground impedance is based on the Delany and Bazley model [8].

Finite size reflectors Fresnel zones contribution is taken into account applying the work of Clay et al. [9].

The atmospheric turbulence model used is based on the work of Nota, Barrels and van Maercke: Harmonoise [10].

The Sound Path Explorer (SPE), a module used by OTL - Terrain, is an in house developed algorithm to detect valid diffraction and reflection sound paths from source to receiver in a proper 3D environment.

Sound path detection is based on the image source method and the Geometrical Theory of Diffraction according to Keller [11].

The following equations respectively represents the calculation result for the ρ statistical reflection coefficient, R_p the plane wave reflection coefficient and the spherical reflection coefficient Q .

$$\rho = 1 - a$$

$$R_p = \frac{Z_m \cos \theta - \rho c}{Z_m \cos \theta + \rho c}$$

$$Q = R_p + (1 - R_p) F(w)$$

α = statistical absorption coefficient,
 Z_m =surface impedance,
 θ =angle of incidence,
 ρc =characteristic impedance,
 $F(w)$: full description on Ref. [4].

5 Trigenerator whole ensemble

Application of the SEA model

5.1 Engine box (container)

The engine box is a typical sea-land container with some Noise control treatment in which Noise emission is expected through walls and from some leaks due to doors sealing and cable passages.

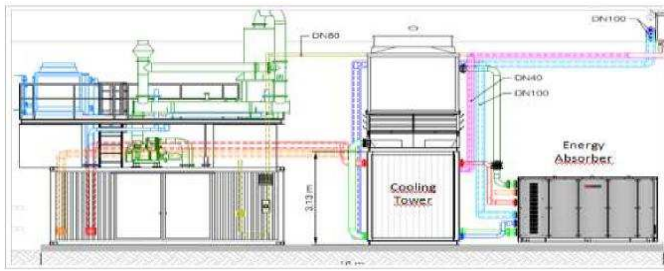


FIGURE 2: TRIGENERATOR ENSEMBLE

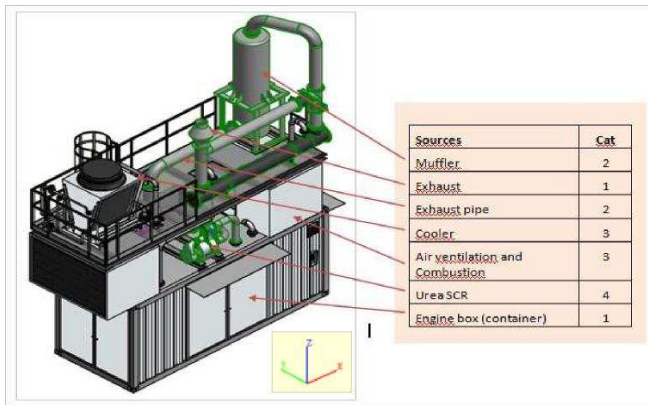


FIGURE 3: TRIGENERATOR MAIN SOURCES

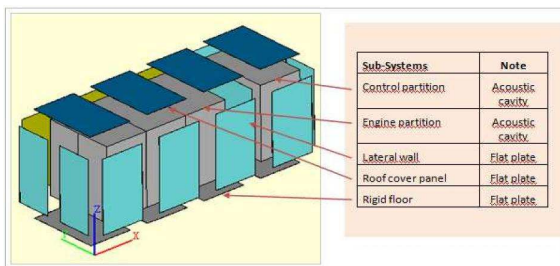


FIGURE 4: SEA MODEL OF THE ENGINE BOX (CONTAINER)

For a SEA model like the present one, we need to define at least two basic acoustic cavities, specifically engine and control room, separated by a steel partition and walls (2 mm thick corrugated steel with damping material), rockwool and perforated steel sheet. Floor is 5mm thick striped sheet, no damping. Roof is 3mm thick steel plate with acoustic coating.

In order to take into account some directivity and the different sources distributed inside the container along the power-line (fan-cooler, engine block, gearbox, generator) the whole container was subdivided in several sub-cavities, pertinent walls and panels as geometrical boundaries.

5.2 Data for running the model

Noise Source "a": Container or Engine box

- The noise data collected from Technical documents of the generator were about the Sound Power determination using ISO 3744 (Lp Sound Pressure Level measurements) in free field over a reflecting floor;
- Cavities take in account of some sound absorption;
- Walls and partition parameters follow the mass-low, modal density and damping behaviour;
- Junctions were left to default CLF (Coupling Loss Factor) between bolted steel plates with foam or rubber sealing;
- A semi infinite fluid field was created up to 10m distance from the container.
- Ensemble of ray-vectors connecting the Point at 10m with each cavity and each wall or partition facing the exterior field.
- A ray-vectors ensemble connecting the point at 10m with each cavity and each wall or partition facing the exterior field was created too.

5.3 Others Sound sources

Noise Source "b": Engine exhaust (chimney)

Engine data-sheet showed chimney mouth Lp at 1 m in 1/1 octave band noise spectra. Sound power was calculated using a 2 m diameter emission sphere.

We estimated its Lp by adding a muffler with a given attenuation and by considering a sphere emission from the chimney at about 6 m height from the ground at distance of 10 m from the container (actually a sphere less a spherical cap), including a 3dB increase of the Lp due to floor reflection; the estimated Lp value reached 67.3 dBA at 10 m distance from container walls in X+ direction.

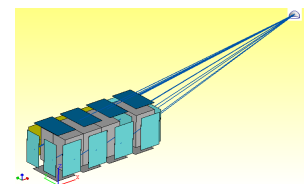


FIGURE 5: CONTAINER SEA MODEL WITH RAYS VECTORS TO RECEIVER IN A SEMI-INFINITE FIELD

Noise source “c”: exhaust piping line

The exhaust gas pipeline is a steel pipe DN150 having a thickness of about 7mm and a diameter of 150mm. Starting from the Sound Power values used for the chimney, we estimated the acoustic transparency of the pipe walls and added some sound insulation on top of standard thermal insulation. See Table 1 for the calculated data which lead to an Lp,A at 10m of 61.8 dBA.

Muffler with a given attenuation and by considering again a sphere emission from the chimney at about 6m height to the ground distance of 10m from the container (actually a sphere less a spherical cap), including a 3dB increase of the Lp due to Floor reflection, we estimated an Lp value of 67.3 dBA at 10m distance from container walls in X+ direction.

1/	Lp,1m chimney	Lw spher	LpA 10m for virtual	TL pipe thic	LpA 10m for	TL with 80mm	TL coib. Term	Lp A a 10
					C1+C2		C	C3+
	Lin	A	7.1	31.1				
31.5	118	79	86	55.5	-13	42.5	-15	40.5
63	125	99	106	75.2	-17	58.2	-22	53.2
125	130	114	121	90.1	-22	68.1	-32	58.1
250	125	116	123	92.2	-27	65.2	-43	49.2
500	126	123	130	99.5	-32	67.5	-45	54.5
1K	122	122	129	98.8	-35	63.8	-45	53.8
2K	114	115	122	91.2	-33	58.2	-45	46.2
4K	104	105	112	81.7	-35	46.7	-45	36.7
8K	98	97	104	73.6	-40	33.6	-45	28.6
16K	95	92	99	68.6	-45	23.6	-45	23.6
Lin	133	127	134	10		72.8		61.8

Table 1: Chimney Noise Calculation sequence

Noise sources “d”: Cooling tower and Energy absorber

The two other sources that form the whole Tri-Gen system are the Cooling Tower and the Energy Absorber as presented in the below drawing, which reports Lp,A lower than 60 dBA at 1m.

6 SEA calculation Results

The following Graph in Figure 5, reports the Sound Power Level (118 dB Lin overall) and the Sound Pressure Levels generated in the cavities and at 10m distance in free field (Lin and A weighted).

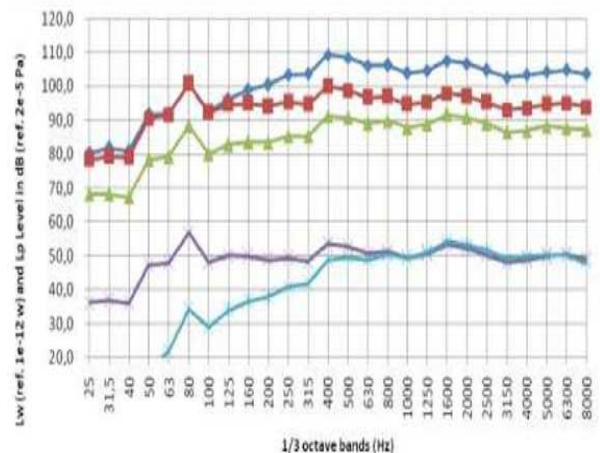


FIGURE 6: SEA CALCULATION RESULTS FOR THE EXTERNAL NOISE IMMISSION AT 10M DISTANCE IN FREE-FIELD: SOUND POWER LEVEL (118 DB LIN OVERALL) AND THE SOUND PRESSURE LEVELS GENERATED IN THE CAVITIES AND AT 10M DISTANCE IN FREE FIELD (LIN AND A WEIGHTED).



7 Application of OTL model for spherical waves propagation

As for the SEA case, we did not consider cooling tower and energy absorber in the acoustic model so the calculation only concerns the co-generator. Theoretical receptors were placed around the Co-Gen installation at 10 m distance at a height of 1.7 m and 3 m respectively above the ground using Free Field simulation.

The considered sources contributions are: a) the engine box (container), b) the chimney mouth and c) the exhaust piping.

In this case a slightly different concept from the SEA sound source modelling has been used as an ensemble of “volumes cavities”, each of them with an internal sound source having a Sound Power value as a fraction of the total power.

The following 3 Figures represent the Propagation model with principal sound sources (top) and volume cavities (middle), receptor is located at 10m distance and 3m height (bottom).

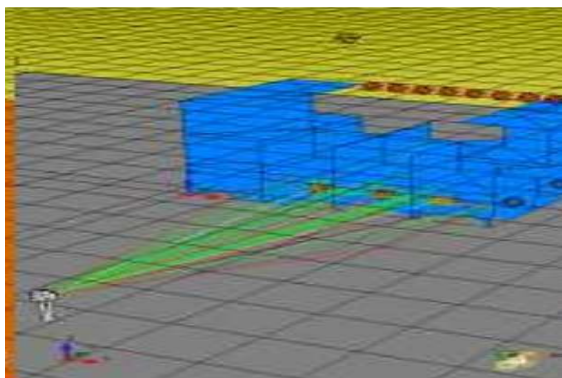


FIGURA 7; PRINCIPAL SOUND SOURCES

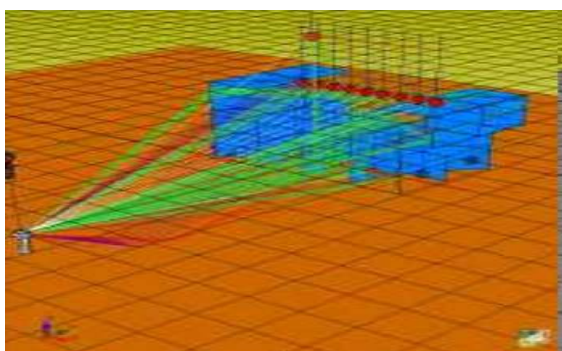


FIGURA 8: VOLUME CAVITIES DETAIL

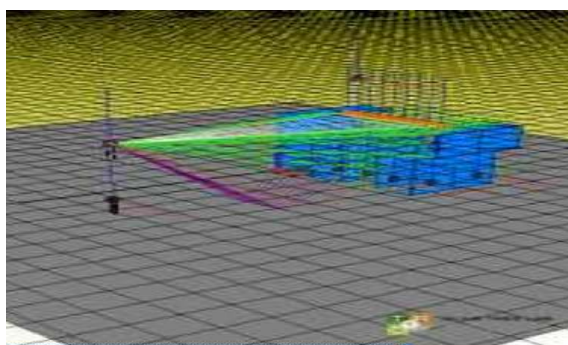


FIGURA 9: RECEPTOR LOCATION

8 Final Results SEA vs, OTL.

Scope of the work was the comparison between SEA and OTL prediction methods. The SEA and OTL calculated L_p 's at receptor location, due to the engine box shows a difference of only 0,1 dBA, for the L_p 's from the chimney end-pipe the is of 0,3 dBA and for the exhaust piping only 0,8 dBA difference. The comparison between SEA and OTL prediction methods show a very good similarity in results, as reported in Table 2..

Source	dBA using SEA	dBA using OTL hybrid
Engine box	62.3	62.2
Chimney end-pipe	67.3	67.0
Exhaust piping	61.8	62.6

Table 2: dBA levels calculated in a position at 10m distance and 1.7m height

9 Conclusion

In the present work we have presented two acoustic modelling methods (SEA and OTL) to estimate noise levels from co-generator and tri-generator ensemble in free field. While the SEA method is based on energies calculation and gives precise results including both acoustic and mechanical excitation of the source structure, the OTL method is an hybrid solution between ray tracing and spherical waves propagation; optimizes the fast calculation of ray-tracing at long distance with the higher precision of spherical waves propagation at a closer distance or in all cases in which the sources cannot be approximated by regular emission form (sphere, cylinders, and so on).

This work is based and exactly matches the previous study as re-reported in Ref. [14].

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