HYBRID OPTIMUM DESIGN OF INDUSTRIAL NOISE CONTROL DEVICES

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Abstract. *The industrial environment is characterized as one of the main sources of sound pollution, that have decisive impact over worker health, as well the comfort and the public calmness. Usually, acceptable noise level over these environments is recommended by standards and legislations. If possible, the noise control must be made in the noise source, using quiet machines or quiet systems. However, for thechnical and economic reasons, exceptionally it becomes possible. In this way, the present work has as main objective to development an hybrid optimum noise control procedure in ducts. The target is to attenuate the noise generated by gases flow an industrial exhaustion system. Both, passives and active thecnics are considered. As passive control both resistive and reactive silencers are applied, always with the possibility of adding an active noise control for attenuation of frequencies below 250Hz.*

Keywords: *Optimal Noise Control Design, Resistive Silencers, Reactive Silencers, Active and Passive Noise Control.*

1. INTRODUCTION

The problem of the noise control in exhaustion systems radiating high sound power levels, in a broad band, presents a wide variety of techniques available to solve it (Snyder, 2000). These techniques can be divided in two main groups: (i) techniques of passive control and (ii) techniques of active control. Particularly, passive techniques can be subdivided in two groups, namely: (ia) acoustic energy flow redirection and (ib) acoustic energy flow reduction.

This work considers application of acoustic energy flow reduction, either modifying the impedance of the duct (reactive techniques), or absorbing the remaining acoustic energy (resistive techniques). In general, reactive techniques are applied to attenuate noise of low frequency (below 250*Hz*), providing more efficient solutions than resistive techniques. On the other hand, resistive techniques, being more economic to be implemented, are well applied to attenuate noise of middle and high frequencies (greater than 250*Hz*).

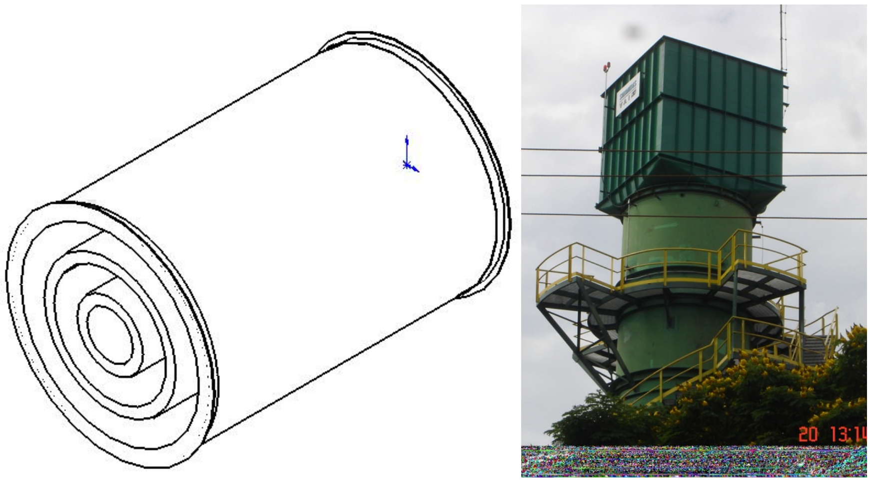
Among the reactive techniques for passive noise control, the most popular has been the side-branch resonator to the exhaustion duct, causing small fluid dynamic energy loss (Gerges, 1992). This type of resonator can be designed to present an optimal performance for low frequencies. Among the resistive techniques, the dissipative silencers with baffles are useful for solving noise control problems involving continuous noise spectrum, such as fan noise. This type of device can be designed to present the best performance in middle and high frequencies.

The present work considers a hybrid optimum noise control procedure for attenuation in a broad band noise, using both, a Helmholtz resonator (Barron, 2003) for low frequencies attenuation and a resistive silencer (Bistafa, 2006) for attenuation of the middle and high frequencies. Additionally, a system of active noise control (Papini and Pinto, 2007) and (Papini, Pinto and Morais, 2008) can be used to complete the hybrid system.

Section 2 is dedicated to the main fundaments of resistive silencer baffle-type. Concepts involved in the design of a Helmholtz resonator are shortly described in section 3. In the sequence, some details of active noise control are briefly commented. The hybrid optimum design is proposed in section 5. A practical example is presented in section 6, based on a real system of a Brazilian industrial plant. The results are very encouraged.

2. RESISTIVE SILENCERS – BAFFLE-TYPE

A baffle-type circular silencer is shown in Figure 2.1. The resistive silencers usually have wide-band noise reduction characteristics and a minimum pressure loss through the silencer. In general, the fall of permissible pressure of these devices meets in the band of 125 to 1500*Pa* (Beranek at. al., 1992). These are devices used in the noise control problems involving continuous noise spectrum, such as fan noise, intake and exhaust noise from gas turbines, noise through access openings in acoustic enclosures, of conditioning, and industrial fans in the cooling towers (Beranek at. al., 1992) and (Barron, 2004). The dissipative silencers attenuate the noise for the conversion of the acoustic energy in thermal energy through the attrition enters rocking gas particles and staple fibers or pores of the absorbent materials.



(a) **(b)**

Figure 2.1 – Baffle-type dissipative silencer: a) with concentric baffles; b) with rectangular baffles.

As it illustrates Figure 2.1, the dissipative silencers consist of a body, rectangular or normally cylindrical, internally coated with absorption material acoustics. Frequently, in its interior are inserted baffles, consisting of the same absorbent material, with the objective of increasing the sonorous absorption, consequently, decreasing the size of the silencer (Gerges, 1992).

Usually, the silencers are made using material of acoustic absorption rock or glass wools covered for fabric and protected by perforated plates or light metallic screens. A basic premise for a good attenuation performance is the resistivity of this absorbent material.

3. REACTIVE SILENCERS – HELMHOLTZ RESONATOR

Figure 3.1 presents a Helmholtz resonator connected laterally to a main duct. The Helmholtz resonators are used to attenuate the noise in low frequency among others that compose the sound spectrum of the air intake of engines of internal combustion, noise of boilers, noise from exhaustion systems of industrial gases and others. This type of device attenuates the noise for the reflection of the incident sound wave to back to noise source and, complementarily, for the dissipation of a small part of the sound energy internally to the resonator (Beranek at. al., 1992).

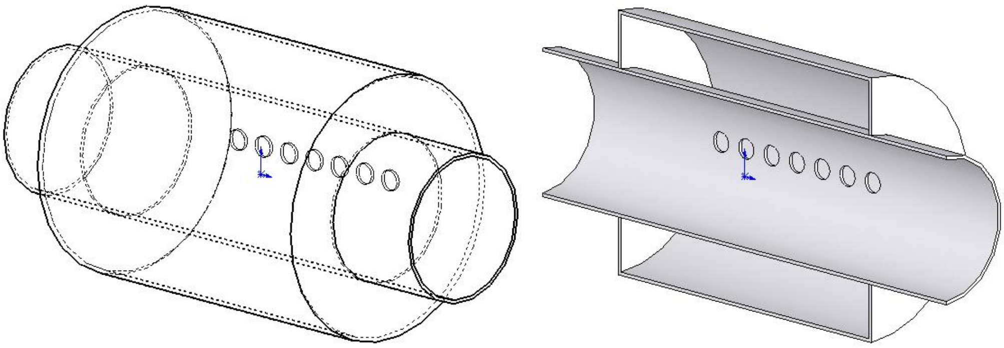


Figure 3.1 – Side-branch resonator.

As it illustrates Figure 3.1, of geometric language, the resonator of Helmholtz consists of a chamber with volume V, connected with a main duct through diameter orifices d.

4. ACTIVE NOISE CONTROL SYSTEM

As alternative to the passive noise control, there is a variety of techniques of active noise control. Now a day, active noise control techniques represents a popular application of technology, especially in USA (Snyder, 2000). A variety of possible ways exists to apply an active noise control system in a duct. Figure 4.1 presents an example of a system of active noise control applied to an exhaustion system of industrial gases. The active noise control is efficient to attenuate low frequencies of noise. This technique is being developed quickly, therefore allows improvement in control of noise with benefits for low size, weight, volume and costs (Snyder, 2000). Currently, the active noise control, in Brazil, is found in development only academic.

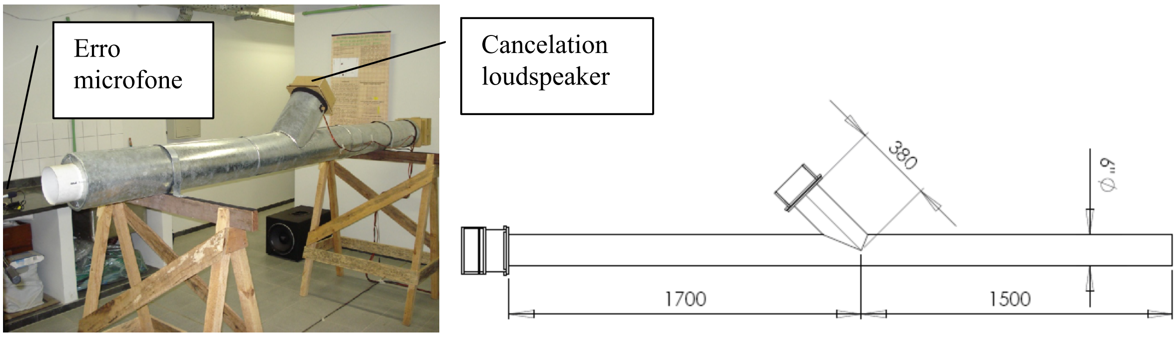


Figure 4.1 – Active noise control

5. PROCEDURE FOR THE OPTIMIZED HYBRID DESIGN

In order to perform the optimized design of noise control problem from a duct of industrial exhaustion system, it is considered a procedure that applies, in accordance with the characteristics of each one, the resistive passive control, the reactive passive control and the active noise control.

The hybrid procedure performance is evaluated by comparing the noise critical level (*NCL)* with the remaining sound pressure level (*SPL)*, *A*-weighted, at 1*m* of the main source (Bistafa, 2006).

This can be understood through the following methodology:

5.1. Global Methodology for Hybrid Design:

In order to implement this optimized hybrid design of a noise control; it carries through the following stages:

- Phase 1 (Starting): for an industrial exhaustion system, it determines the basic parameters for the design of the noise control system (as description ahead).
- Phase 2 (Design of the reactive system): to attenuate the low frequencies is projected Helmholtz resonator.
- Phase 3 (Design of the resistive system): to attenuate middles and high frequencies is projected a baffle-type silencer, to be placed in sequence to the Helmholtz resonators.
- Phase 4 (Design of the active system): to attenuate low frequencies which passive control is limited to dissipate, considering all restrictions, for example: the available length duct.
- Phase 5 (Finishing): it complements the project of the system of noise control considering practical aspects, constructive details of the problem.

5.2. Development of Phase 1: Initialization.

To develop Phase 1, consider the following parameter specification and preliminary steps:

For *Noise Source Identification*:

Option 1 (*noise source in operation*): calculates the sound power levels (*NWS*) through measurements in field:

1. using microphone and applying a method of engineering through standard ISO 3744;
2. using sound intensity probe (in free field), because sound pressure level measurements by itself do not locate the primary noise.

Option 2 (*noise source in design phase*): calculates the sound power levels through the empiric mathematic model.

For *fan* characterization, specify:

i. Fan type.
ii. Rotation of the rotor (*n*).
iii. Blade number of the rotor (*k*).
iv. Pressure increment (ΔP).
v. Mechanical power (*W*).

For *exhaustion duct* characterization, specify:

vi. Geometric of the exhaust duct (circular or rectangular).
vii. Internal dimensions: width or diameter.
viii. Duct wall thickness (*e*).
ix. Available duct length ($L_{available}$).

For *flow* characterization, specify:

x. Flow fluid design temperature (T_P).
xi. Flow fluid operation temperature (T_{op}).
xii. Fluid volumetric flow (*Q*).
xiii. Fluid specific mass (ρ).
xiv. Fluid molecular mass (*M*).

For *measurements* characterization, specify (only for operating systems):

xv. Ambient temperature (T_0).
xvi. Air relative humidity (*U%*).
xvii. Distance between the measurement sensor and the source.
xviii. Free or reverberant field?
xix. Wind speed.

5.3. Development of Phase 2: Design of Helmholtz Resonator.

To develop Phase 2, consider the following algorithm:

Algorithm for the Design of a System of Helmholtz Resonators:

Step 0: (initialization)

0a Determine the propagation sound speed *[c (m/s)]*.
0b Specify the specific acoustic resistance for wire screen $[R_{screen}\ (N\ s/m^3)]$.
0c Specify the A-weighted factor based on normalized frequencies.
0d Specify the noise critical level (*NCL*), A-weighted, at 1*m* at sound source to be treated.
0e Determine the blade passage frequency of the fan:

$$BPF = \frac{n\ k}{60} \tag{5.1}$$

0f Identify the sound power levels radiated by the noise source $[NWS_j\ (\ j = 1, \dots, N\ )]$.
0g Identify the noise frequencies lower than 250*Hz* to be treated $[\ f_{0_i}\ \ (i = 1, \dots, N\ )]$.
0h Make $i = 1$.

Step 1: for the frequency f_{0_i} :

1a Define the frequencies f_1 and f_2, around f_{0_i} based on normalized frequencies (1/*n* octave).
1b Define the maximum transmission loss for the silencer, $PT_0\ (dB)$.
1c Calculate the resonance curve form factor β (Barron, 2003), through the following equation:

$$\beta = 10^{PT_0/20}, \tag{5.2}$$

1d Calculate the acoustic resistance required for the resonator, $R_A\ [Pa\text{-}s/m^3]$, through the equation:

$$R_A = \frac{\rho_0\ c}{2(\beta - 1)S_{duct}}, \tag{5.3}$$

where, S_{duct} denotes the main duct transversal section, ρ_0 the specific mass of the fluid and c the sound velocity of flow fluid inside the main duct.

1e Calculate the acoustic quality factor, Q_A (dimensionless), through the following equation:

$$Q_A = \frac{f_0\,\beta}{\left(\beta^2 - 2\right)^{1/2}\left(f_2 - f_1\right)} \qquad 5.4$$

1f Calculate the acoustic mass in the neck, M_A $[kg/m^4]$, through the following equation:

$$M_A = \frac{Q_A\,R_A}{2\pi\,f_0} \qquad 5.5$$

1g Calculate the acoustic compliance related to the volume of the chamber and the acoustic mass, C_A $[m^3/Pa]$, through the following equation:

$$C_A = \frac{1}{4\pi^2\,f_0^2\,M_A} \qquad 5.6$$

1h Calculate the volume of the chamber of the resonator, V $[m^3]$, through the following equation:

$$V = C_A\,\rho_0\,c^2 \qquad 5.7$$

1i Calculate the additional specific acoustic resistance, caused by wire screens (if wire screens will be used), (Barron, 2003):

$$\Delta R_A = \frac{N_{screen} R_{screen}}{\pi\,a^2\,N_{roles}}\,, \qquad 5.8$$

where, N_{screen} represents the number of wire layers, a the ray of orifice and N_{roles} the number of roles.

1j Calculate the global resistance of the resonator by the equation:

$$R_{disp} = R_A + \Delta R_A\,, \qquad 5.9$$

where $\Delta R_A = 0$ if wire screens are not used.

1k Calculate the optimized role radius, through the following equation:

$$r_{roles} = \frac{3\pi}{16}\left[\frac{M_A\,R_{available}}{R_A} - e\right] \qquad 5.10$$

1l Calculate the number of roles, through the following equation:

$$N_{roles} = \frac{R_{available}}{\pi\,r_{roles}^2\,R_A} \qquad 5.11$$

1m For the resonance frequency f_{0_i}, calculate the resonator performance for all frequencies to be attenuated, through the following equation:

$$IL_{i,j} = 10\log\left(\frac{\beta^2 + Q_A^2\left[\left(f_{0j}/f_{0_i}\right) - \left(f_{0i}/f_{0_j}\right)\right]^2}{1 + Q_A^2\left[\left(f_{0_j}/f_{0_i}\right) - \left(f_{0i}/f_{o_j}\right)\right]^2}\right)\ [dB] \qquad j = 1,\ldots,N. \qquad 5.12$$

Step 2: Update the remaining sound power level:

$$SWL_{rem_j} \leftarrow SWL_{rem_i} - IL_{i,j} \quad [dB] \qquad , \ j = 1,\dots,N \qquad 5.13$$

Step 3: If $i = N$, go to Step 4, otherwise it makes $i = i + 1$ and go back to Step 1.

Step 4: Calculate the remaining sound pressure level, evaluated at $1m$ of the exit section of the device noise control (for free field with floor), (Bistafa, 2006):

$$SPL_{rem_j}(r,\theta) = SWL_{rem_j} - 20\,log(r) + DI_\theta - 10\,log\left(\frac{\Omega}{4\pi}\right) - 11 \quad [dB] \qquad , \ j = 1,\dots,N \qquad 5.14$$

where, r denotes the distance between the noise source and the receiver, DI_θ denotes the sound source directivity, and $\Omega = 2\pi$ if the receiver is near the floor, or $\Omega = 4\pi$ if the receiver is in free field.

Step 5: Calculate the SPL_{rem_j} $[dBA]$.

Step 6: Calculate the SPL_{global} $[dBA]$ using SPL_{rem_j} $[dBA]$.

Step 7: Compare SPL_{global} with NCL $[dBA]$. If $SPL_{global} \le NCL\ [dBA]$, then finish the project, otherwise initiate the design of the resistive silencer.

5.4. Development of Phase 3: Optimum Resistive Silencer.

To develop Phase 3, consider the following algorithm:

Algorithm for the Resistive Silencer Design:

Step 8: (initialization)

8a Identify the remaining sound power level radiated, SWL_{rem_j} $(j = 1,\dots,N)$.
8b Identify the frequencies greater than $250Hz$ to be treated, f_j $(j = 1,\dots,N)$.
8c Define the absorbent material and the respective coefficients of absorption (Bistafa, 2006).
8d Exclude the frequencies which absorption coefficients are equal zero (non treatable frequencies).
8e Make $j = 1$.

Step 9: to obtain an optimum design:

9a Calculate the flow thermodynamic and aerodynamic parameters.
9b Set the maximum flow velocity (V_{max}) in the free section of the silencer as 20 m/s (Gerges, 1992).
9c Define the volumetric outflow Q_V (m^3/s).
9d Calculate the cylindrical free internal area of the resistive silencer, through the equation:

$$A_{int} = \frac{Q_V}{V_{max}} \qquad [m^2] \qquad 5.15$$

9e Calculate the absorbent perimeter (P_{int}) according to the current geometry.
9f Calculate the maximum available length for resistive silencer:

$$L_{max} = L_{available} - L_{react} \qquad 5.16$$

9g Find the maximum value of L_{resist} such that:

$$L_{opt} \leq L_{max} \qquad 5.17$$

$$SPL_{global} \leq NCL$$

where,

$$SPL_{global} = 10\, log\left\{ \sum 10^{\left[\frac{SPL_{rem_j}(L) - A_j}{10}\right]} \right\} \; [dBA] \qquad j = 1, \ldots, N \qquad 5.18$$

where A_j represents the A-weighted factor and

$$SPL_{rem_j}(L) = SWL_{rem_j} - IL_j(L) - 20\, log(r) + DI_\theta - 10\, log\left(\frac{\Omega}{4\pi}\right) - 11 \quad [dB] \qquad j = 1, \ldots, N \qquad 5.19$$

where,

$$IL_j(L) = 1{,}05\, \alpha_j^{1,4}\, L \frac{P_{int}}{A_{int}} \quad [dB] \qquad j = 1, \ldots, N \qquad 5.20$$

5.5. Development of Phase 4: Optimized Design of an Active Noise Control System.

Step 10: If $L_{opt} < L_{max}$, the passive control is sufficient to attenuate the noise. Then go to Step 12 to finish the hybrid design. Otherwise, goes to Step 11.

Step 11: Check the global sound pressure level (SPL_{global}). If $SPL_{global} \leq NCL$, the passive control is sufficient to attenuate the noise. Then, go to Step 12 to finish the hybrid design. Otherwise, improve the treatment of the frequencies near to 250*Hz* by using an active noise control system. For that, use (Papini and Pinto, 2007) and (Papini, Pinto and Morais, 2008) with other bibliographies, if necessary. After goes to Step 12.

5.6. Accomplishment of Phase 5: Finishing.

Step 12: Complements the project of the noise control system considering economic and practical aspects, constructive details or specificities of the problem.

6. RESULTS

For the acoustic treatment of the high sound power levels radiated from the fan of the exhaustion system of a steel industry, the hybrid optimum design procedure will be applied.

6.1 Dates used to initialize the design:

Fan Specifications:
Type: *centrifugal*;
Rotation control: *constant velocity*;
Design temperature: 60^0C;
Rotation: *1780rpm;*
Blade passing frequency: *267Hz;*

Flow Characteristics:
Barometric pressure: *101,325kPa*;
Fluid: *air*;

Volumetric flow: *320000*m^3/h;
Static pressure of the fan: *413mmca*;
Operation temperature: 60^0C;
Mass specific of the fluid: *1,059*kg/m^3.

For *exhaustion duct* characterization, specify:
Geometric of the exhaust duct: *rectangula*r
Internal dimensions of main duct: *2000 x 1400mm*.
Duct wall thickness: *6,35mm*.
Maximum available duct length: *3000mm*

Predicted *Sound Power Levels* (Bistafa, 2006):

Frequency (Hz)	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
***SWL* (*dB*)**	115	109	120	109	107	102	100	99

The sound power levels (*SWL*) was estimated at fan exit.

6.2 Results of Hybrid Optimum Design:

The exhaustion system schematic drawing, including the noise control system designed by applying the hybrid optimum design algorithm is shown in Figure 6.1.

The controlled plant has two Helmholtz resonators and a baffle-type resistive silencer.

Table 6.1 and Table 6.2 present the optimized dimensions for the first and for the second Helmholtz resonator, respectively.

Table 6.1 – Dimensions of the first Helmholtz resonator.

Narrow band	50 to 80*Hz*
Volume (m^3)	3,72
Diameter of roles (*mm*)	85,29
Number of roles	60
Number of screen	1

Table 6.2 – Dimensions of the second Helmholtz resonator.

Narrow band	100 to 160*Hz*
Volume (m^3)	2,88
Diameter of roles (*mm*)	36,02
Number of roles	481
Number of screen	1

The first resonator was dimensioned to attenuate the narrow band between 50 and 80*Hz*. The second one was dimensioned to attenuate the narrow band between 100 and 160*Hz.*

The remaining sound power was attenuated by the resistive silencer. Table 6.3 presents the optimized dimensions for resistive silencer. The solution use baffles with 50*mm* thickness and wool fiber glass as absorbent material.

Table 6.3 – Dimensions of the resistive silencer.

Wide band	125 to 8000*Hz*
Baffles number	10
Length (*mm*)	2728
Internal diameter *(mm*)	3379

The three devices had been mounted sequentially. The critical noise level defined for this project was 75*dBA* at 1*m* sound source.

The optimum length obtained (L_{opt}*=2728mm)* is less than the maximum allowed length *(3000mm)*. Then the passive noise control system is able to attenuate the noise adequately. As consequence, the active noise control it is not necessary for that system.

In despite of, a complementary active noise system has been suggested by the authors to the steel industry. In fact, to attenuate noise that can eventually happen in consequence of particular operating conditions was suggested an auxiliary active system to increase the security by a device to be activated only in special conditions.

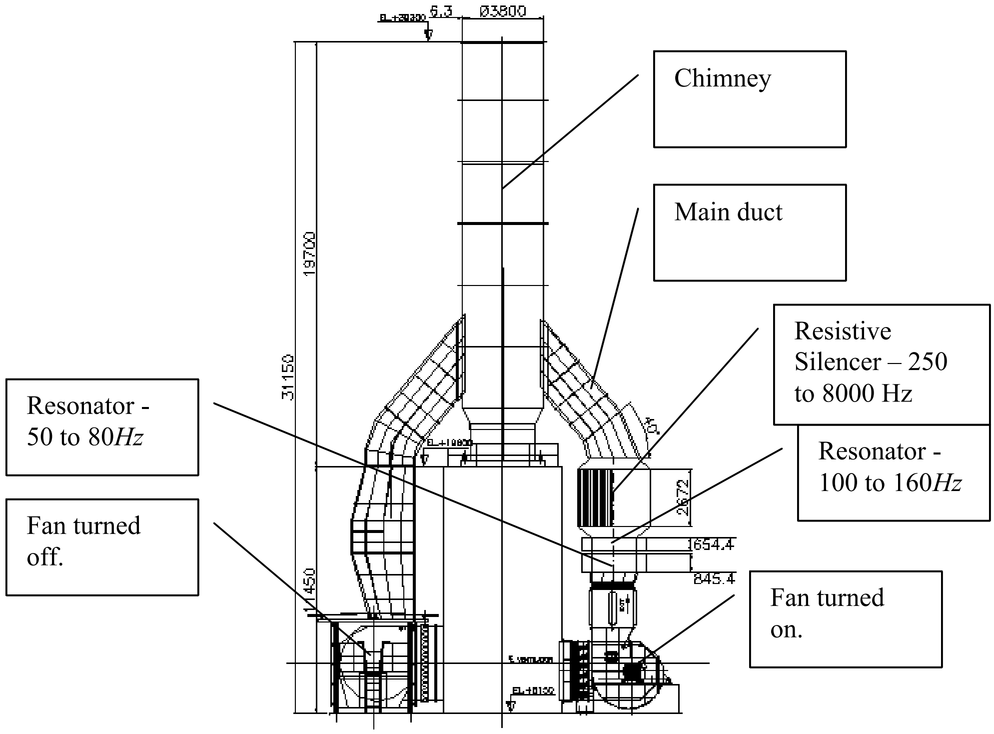


Figura 6.1 – Exhaustion system with optimum noise control system.

Table 6.4 presents the performance of the hybrid optimum design. The results for remaining sound power level are shown in octave band. For remaining sound pressure level *A*-weighted the results are shown in octave band and global level. The global sound pressure level at 1*m* is 75*dBA*.

Table 6.4 – Performance of the hybrid optimum design.

	Frequency (*Hz*)	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**Global (*dBA*)**
***SWL*exhaution (*dB*)**		115	109	120	109	107	102	100	99	
Resonator 50 to 80*Hz*	*IL* (*dB*)	9	2	0	0	0	0	0	0	
	*SWL*remaining (*dB*)	106	107	120	109	107	102	100	99	
Resonator 100 to 160*Hz*	*IL* (*dB*)	3	11	3	1	0	0	0	0	
	*SWL*remaining (*dB*)	103	96	117	108	107	102	100	99	
Resistive Silencer	*IL* (*dB*)	0	3	27	38	41	33	38	38	
	*SWL*remaining (*dB*)	103	93	90	70	66	69	62	61	
***SPL*remaining at 1*m* (*dBA*)**		69	70	73	59	57	61	55	54	**75**

7. CONCLUSIONS

A hybrid optimum design methodology was proposed for noise control systems of industrial exhaustion ducts. The procedure considers the possibility of using simultaneously Helmholtz resonator, dissipative silencer and active noise control systems.

An algorithm was proposed to use the theoretic and practical concepts in a well organized way.

A practical example of a steel industry plant was presented were the optimum design attends the standards dispensing the use of active noise control system. Only for additional considerations authors suggest the use of active control to increase the security on kept legal requirements.

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