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Hybrid approach to noise control of industrial exhaust systems

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ABSTRACT

This study aims to present a procedure for optimal design of hybrid noise control systems for exhaust ducts from large fans and high radiated sound power, in an industrial environment. Reactive, resistive or hybrid silencers are considered as alternatives with the possibility of adding a supplementary active noise control system to mitigate remaining noise of multiple low frequencies, which are difficult to be attenuated by passive means.

Algorithms are proposed for the implementation of optimal hybrid passive/active design and the main theoretical and practical fundamentals involved are showed.

Two effective solutions for a real industrial exhaust system are presented as examples of applications of hybrid optimal design procedure proposed. The first one uses of Helmholtz resonators (to attenuate lower frequencies) and resistive silencer parallel lamellae (to lower the midrange and high frequencies). The second one adopts resistive silencer of parallel lamellae and a forward active noise control system.

This is a hybrid approach that is not treated in this manner in the available literature, especially in the text books of applied acoustics engineering.

1. Introduction

In industrial plants it is common the existence of exhaust gas process systems that generates noise above the limits set by environmental and labor standards. Conceptually, the most effective way of reducing industrial noise is through direct intervention on the sound source [7]. However, this alternative, in most cases, is not feasible in the existing industrial plants, requiring, as a solution, interventions in the means of propagation of sound energy. Examples of such interventions are: (i) total or partial enclosure of the sound source; (ii) inserting acoustic barriers between the sound source and receivers; (iii) installation of acoustic absorbers at the workplace; (iv) installation of silencers in ducts of exhaust systems. All these possibilities mentioned are classified as passive control techniques [1,5,6,7]. More recently, active control systems have also been considered as an alternative [12].

In this study, specifically, it is analyzed the noise attenuation system design applicable to industrial chimneys, considering both the passive type and the active type.

According to [1,4–7,11,12] passive technical noise control in industrial chimneys is subdivided into two categories: those that absorb sound energy (resistive passive techniques); those that alter the impedance of the duct (reactive passive techniques).

The active techniques, based on the emission of sound waves canceling, represent a more recent alternative with great potential of industrial application [6]. There are many scientific studies on active noise control that are targeted to industrial fans, exhaust automotive engine and passenger compartments of cars, trucks or airplanes [8].

In the bibliography on the subject [1,4–7], the alternative for the treatment of noise in industrial ducts have been presented exclusively, i.e., it should be chosen one of the possible techniques, not addressing the simultaneous application of more than one.

Differently, this paper proposes a hybrid approach to noise control system design by combining different passive techniques with an active technique when necessary.

In the present work the analysis of resistive silencers is based on empirical formula of Sabine [5,7,12]. Regarding the reactive silencers, the Helmholtz resonators are considered [1], adopting an approach by concentrated parameter, based on analogy with mechanical or electrical systems.

For the active control a forward system using FXLMS algorithm in the time domain is approached, controlling the potential energy density [9]. The FXLMS algorithm is relatively simple compared to other active noise control methods, easy to implement and provides satisfactory performance in simple applications, particularly those where the sound propagation occurs through plane waves [7].

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The presentation of the hybrid approach for the realization of the project is enriched with flowcharts and sequenced steps with criteria for the selection and design of each of the noise control system.

To illustrate, in a practical way, the approaches studied alternative solutions to a real problem of a steel plant for the manufacture of steel rebars, at the state of Minas Gerais, Brazil, are presented, which makes use of a 1525 kW absorbed power fan, generating sound spectrum with relevant loudness between 125 and 8000 Hz. The results found with the proposed solutions are compared with the ones given by a traditional solution, taking into account not only the technical performance, but also manufacturing costs based on updated budgets.

2. Resistives silencers

The resistive or dissipative silencers consist of a duct, typically circular or rectangular, internally lined with sound absorption material. Often, the ducts have concentric or parallel lamellae, made up of the same absorbent material inserted into the duct. Fig. 2.1 shows examples of resistive attenuators with rectangular and concentric lamellae. These devices are widely used on noise control of dust removal systems, air conditioning systems, air intake systems, diffusers of cooling towers, in ducts connected to industrial fans [1,4,6,11]. Also, they are used for the admission and exhaustion of ambient air for cooling groups of motors and generators, hydraulic units and enclosed turbochargers groups in metal cabinets or rooms in masonry [10].

The resistive silencers attenuate noise by converting the acoustic energy into heat energy through friction between the oscillating gas particles and the fibers or pores of the absorbent material. Generally, to achieve a satisfactory level of sound attenuation, the length of the internal treatment should be ten times larger than the diameter of the open duct section [6]. However, the presence of lamellae increases the area for sound absorption, enabling a more compact device in terms of length [5].

To determine the level of sound attenuation when designing an acoustic attenuator resistive type are essential: (i) the acoustic resistivity of the absorbing material; (ii) the thickness of the absorbent material towards the open portion of the duct section (usually recommended to a thickness equal to 1/5 of the radius of the open section of the duct); (iii) the area of sound absorption installed in the duct wall.

The resistivity of the material depends on the flow resistance, which characterizes the pressure difference observed on both sides of an acoustic material [5].

In general, the efficiency calculation of a resistive silencer is based on a parameter called "transmission loss", calculated for each frequency of interest.

2.1. Calculation of transmission losses

In the present work, the transmission losses (PT_i) will be calculated using the empirical formula of Sabine [5,7,12]:

$$PT_i = 1.05 \times (\alpha_i)^{1.4} \times L_{silencer} \times \left(\frac{P_{\mathrm{int}}}{A_{\mathrm{int}}}\right) \quad [\mathrm{dB}], \tag{1}$$

where α_i denotes the sound absorption coefficient (dimensionless) normalized to the i-th octave band, L_{silencer} the silencer length, P_{int} denotes the inside perimeter of absorption and A_{int} denotes the free internal area (wet area).

Eq. (1) takes into account the effects of reflected sound waves and transverse sound waves propagating in the main duct at the inlet and outlet silencer.

In order to use Eq. (1) in ducts with rectangular geometry and provide accuracy equal to or less than 10%, the following restrictions [5] must be respected:

1. each side of the rectangular section (wet section) should have a length between 150 mm and 450 mm;
2. the ratio between the sides of the rectangle should be between 0.5 and 2;
3. the air flow velocity must be less than or equal to 20 m/s;
4. every absorption coefficient (α_i) must be less than or equal to 0.8.

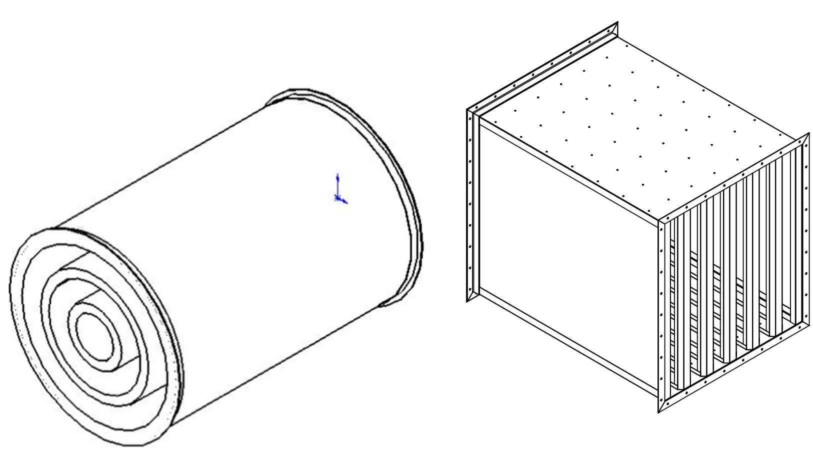
The geometric constraints of (1) and (2) exist as a result of the proportion limits experienced during the determination of the empirical formula [5].

The item (3) refers to a practical upper limit for efficient operation of the insulation material [5].

Finally, the item (4) refers to a practical upper limit for alpha, considering a random impact on the absorbent material. Some authors [4,5] recommend the value 0.8 as the upper limit, even if the value indicated by the manufacturer is greater than 0.8.

3. Reactive silencer – Helmholtz resonator

Fig. 3.1 shows a Helmholtz resonator coupled to a main duct (inner cylindrical section). Basically, this type of resonator consists of a chamber with volume V, communicating with a main duct



(a) (b)

Fig. 2.1. Resistive silencers type lamellae: (a) concentric; (b) rectangular.