

EXPERIMENTAL APPROACHES TO REDUCE NOISE AT SOURCE FOR A CENTRIFUGAL COMPRESSOR

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Compresoarele centrifugale utilizate în industrie reprezintă surse importante de poluare fonică, în special atunci când acestea sunt amplasate în apropierea zonelor rezidențiale.

Scopul acestei lucrări este de a prezenta experimentele și rezultatele obținute în reducerea zgomotului la sursă pentru un compresor centrifugal. Originalitatea lucrării reiese din modul de implementare a cavităților rezonante în geometria compresorului centrifugal, precum și analiza de post-procesare a datelor obținute prin efectuarea măsurărilor acustice înainte și după modificarea compresorului.

La sfârșitul lucrării sunt fi oferite concluzii și este întocmit un plan de acțiuni viitoare aplicabile pentru a reduce un domeniu mai larg de frecvențe.

Centrifugal compressors used in industry are important noise sources, especially when they are located near residential areas.

The purpose of this paper is to present the experiments performed and results obtained to reduce noise at source for a centrifugal compressor.

The ways to implement the resonant cavities in the centrifugal compressor geometry and the post-processing analysis of data obtained by performing acoustic measurements before and after remodelling the compressor represents the originality of the paper.

Conclusions and future action plan to reduce noise on a wider spectrum are offered at the end of the paper.

Key words: noise, source, centrifugal compressor, resonators, reduction

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1. Introduction

Centrifugal compressors from industrial systems use mechanical energy to compress the working fluid. It is obvious that a small fraction of this energy is converted into acoustic energy, propagating in the whole system and manifesting by noise, both locally and in the vicinity of the compression stations [1].

The implementation of noise protection measures has begun since the mid 70's and in the recent years noise protection programs have been applied worldwide in the densely populated area from the vicinity of the compression and transportation station. In our days a fierce fight to reduce turbo-compressors noise and especially the centrifugal compressor noise is carried out by using passive techniques (porous absorbing panels, resonators cavities) or active techniques (generation of noise signals in opposite phases to those generated by noise source). In the field of industrial centrifugal compressors operating near human communities, more and more manufacturers of such equipments are making efforts to reduce the noise generated by them. Among them, we cite the most active: Dresser-Rand, General Electric, Nuovo Pignone. Their reports mention a compressor noise reductions in the range of 10-15 dB(A).

On national level in centrifugal compressor noise reduction we can mention COMOTI-Romanian Research and Development Institute for Gas Turbines which have a 25 years experience in design, production and improvement of centrifugal compressors both for industrial and aviation applications.

The main method chosen to reduce noise is the noise parallel separator system of walls, which are composed by sound absorbing panels along the inlet and exhaust pipes. This passive system relies on the sound absorption sandwich material disposed between perforated walls. The literature does not recommend a certain configuration for sound absorbent panels. Their number and dimensions influences the working fluid speed inside the attenuator. This kind of silencers requires periodic maintenance and replacement, especially at the exhaust [2], [3], [4]. In literature we find only one case of reducing noise at centrifugal compressor using resonator arrays, but without mentioning on how to design them or where to be implemented [5], [6], [7].

Active control is also widely used. With this system an annoying tone can be cancelled by introducing a second one which has 180 degrees opposite phase tone.

Noise reduction at source is proven to be a long and complex method that involves a fundamental research stage and laboratory experiments stage, followed by redesign, prototype design and production of an experimental model to be tested on a test bench. In case of successful outcomes, the system is simplified

and it is not necessary to involve other insulation methods that would assume additional costs and complicate the maintenance.

To develop a noise reduction plan at source for a centrifugal compressor we have established the following objectives:

- Centrifugal compressor noise measurement performed for an experimental model and in order to obtain the fundamental frequency and its amplitude;
- Theoretical calculation and experimental tests to choose optimal dimensions of the acoustic resonators to be implemented in centrifugal compressor geometry;
- Modification of the compressor with the optimal resonators studied in the laboratory;
- Testing of the modified centrifugal compressor on a test bench;
- Comparative analysis of the obtained results.

The paper will presents all these steps and future actions to reduce noise on a wider spectrum.

2. Theoretical aspects regarding the use of acoustic resonators

The optimum design of resonators involves several requirements such as: acoustic, aerodynamic, geometric, mechanical and material and economic requirements.

The Helmholtz resonator and the analogous mechanical and electrical systems are shown in Fig. 1, [8]. The system consists of an acoustic resistance R_A , an acoustic mass M_A and an acoustic compliance C_A . The elements are in “series”, such that the volumetric flow rate is the same for each element. In the acoustic system, the pressure is analogous to the mechanical force or the electrical voltage and the volumetric flow rate is analogous to the velocity of a mass or the electrical current.

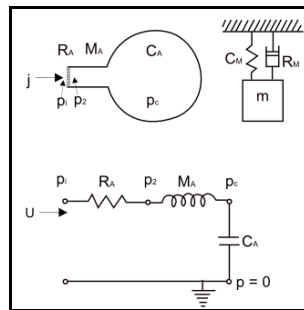


Fig. 1. Equivalent circuit diagram for the Helmholtz resonator. The pressure at one terminal of the acoustic compliance is atmospheric pressure, or the acoustic pressure is zero. The acoustic pressure at the other end of the compliance terminal is p_c . The Helmholtz resonator is analogous to the mechanical system with a spring, damper and mass.

The resonant frequency for the Helmholtz resonator is similar in principle to the natural frequency for vibration of a spring-mass system. For an electrical system, resonance occurs at a frequency such that the electrical reactance is zero. The resonant frequency for the Helmholtz resonator is given by the following expression:

$$f_0 = \frac{1}{2\pi\sqrt{M_A C_A}} \quad (1)$$

This relationship may be used for design purposes to determine the resonator frequency for given resonator dimensions or the required dimensions may be found to achieve a given resonant frequency.

The side-branch muffler is one type of silencer used to reduce noise emission in a restricted frequency range from a mechanical system. The side-branch muffler consists of a Helmholtz resonator connected to the main tube through which the sound is transmitted. The silencer acts to reduce sound transmission primarily by reflecting acoustic energy back to the source, so it is classified as a reactive silencer; however, some energy is dissipated within the acoustic resistance element of the silencer. Some typical configurations for side-branch muffler are shown in Fig. 2.

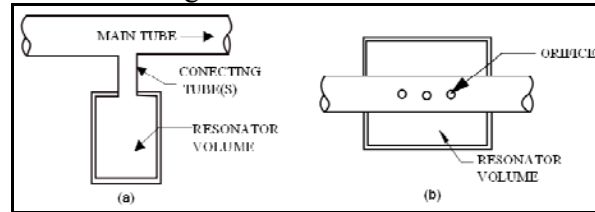


Fig.2. Configurations for side-branch mufflers: (a) resonator connected by a tube or tubes and (b) resonator connected through orifices

In many design situations, the following parameters are known or specified: (a) minimum acceptable transmission loss, TL_{min} ; (b) primary range of frequencies for the silencer operation, f_1 and f_2 ; (c) cross-section area of the main tube, S ; and (d) the type of gas in the silencer, so that the density ρ_0 and speed of sound c are known. There are often more “unknowns” than design equations; therefore, there are several possible solutions for a particular set of design parameters. The design equations may be organized, however, in a form that allows a more direct design procedure. In the following this directed design procedure for a side-branch muffler is outlined.

Practical choices must be made for two of the following quantities to completely specify the design: specific acoustic resistance, R_S ; number of side-branch tubes, N ; side-branch tube radius, a ; resonator volume, V ; and length of the side-branch tube, L . The remaining three quantities may be calculated from the

previously determined values of the acoustic mass, M_A ; the acoustic resistance, R_A ; and the acoustic compliance, C_A .

For a side-branch muffler with an attached resonator volume, the muffler operates quite well around the single resonant frequency of the resonator volume. A long tube (closed tube), Fig. 3, however, has, theoretically, an infinite number of resonant frequencies.

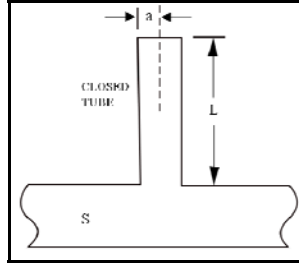


Fig. 3. Closed tube of radius and length as a side branch

In many design cases, additional acoustic resistance, in the form of screens and other elements, must be added to achieve the required total acoustic resistance [9] [10].

3. Experimental work

In order to reduce noise at source for a centrifugal compressor and to apply the theoretical studies, we followed two steps. Then measurements at a centrifugal compressor and calculatus and experimental verification of resonator models for the fundamental frequency identified through measurements.

3.1. In-situ acoustic measurements for a centrifugal compressor

The measurement objective was to identify the centrifugal compressor frequencies for normal operating condition. To achieve this objective, we have used the intensity probe which offered an image of the high acoustic intensity on the compressor case (Fig. 4).



Fig. 4. Measurements for acoustic intensity to perform the compressor case mapping (49 grids surfaces $\sim 18 \times 18 \text{ cm}$ each with a distance between them and case of 18 cm)

The noise intensity measurements were performed according to SR EN ISO 9614-2/2000 “Determining the sound power levels using acoustic intensity. Part 2 – Measurements by scanning.”

In the following, are presented the configuration and preparation steps made to perform noise intensity measurements:

- Establishing the interest area (compressor case);
- Establishing the grids dimensions (49 surfaces, each of them having $18 \times 18 \text{ cm} \pm 0.5 \text{ cm}$);
- Establishing the distance between the grids surfaces and noise source (18 cm away from the case);
- Configuring the measurements chain and calibrating the intensity probe;
- Performing measurements by scanning in 2 cycles for each surface from the grid;
- Processing the results to obtain the global sound power;
- Processing the results for mapping the noise intensity levels.

In Fig. 5 are presented:

- Global sound power level, A weighting frequency analysis resulted from the 49 grid surfaces;
- Global noise intensity level, frequency analysis resulted from the 49 grid surfaces;
- Global sound pressure level, frequency analysis.

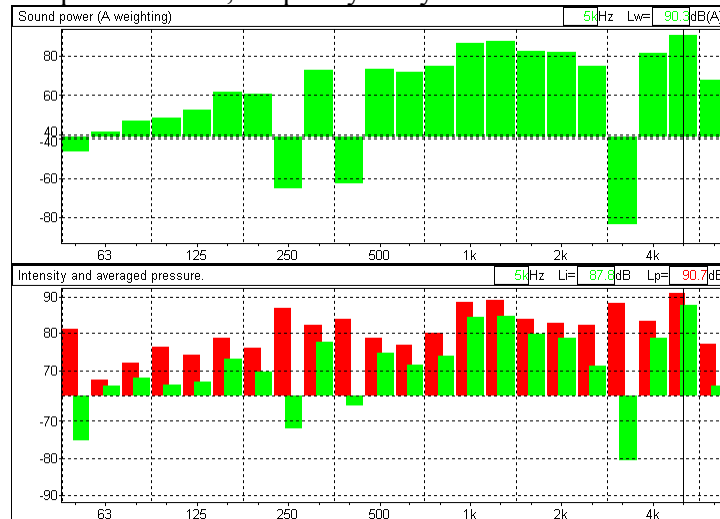


Fig. 5. Noise results obtained by scanning the compressor case surface with the intensity probe

By correlating the results we can outline the pulsation resulted from the compressor rotor at 5 kHz central frequency which is the highest level of the 1/3 octave spectrum.

3.2. Laboratory experiments with 31 resonators models

Next step was to design resonators which are acting for the central frequency of 5 kHz at a normal operating condition for the compressor. We choose to implement the resonators in the centrifugal compressor geometry in the frame of the stator. This idea was also applied at international level [11], but the details of how they implemented the resonators arrays are not given. Furthermore, the mathematical models are only for guidance and for classic examples.

To choose the optimum resonators, we calculated and experimented in laboratory 31 resonator models.

Taking into account the theoretical aspects, we established the inputs for closed tube resonators to be implemented in the EEISG centrifugal compressor [12]. All the resonators were experimented in rectangular channels with 40x30 mm.

The laboratory experiments were performed for different inner radius of the resonant tube and for different cavity geometry. In total we have experimented 10 different resonators models (both Helmholtz and closed tube) with 3 different inner radii: 4 mm, 6 mm and 8 mm, presented in figure 6.

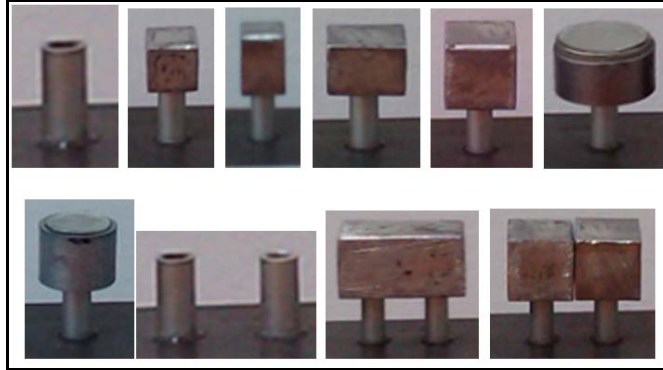


Fig. 6. Resonator models experimented in the laboratory

All the models were made from metal and glued with a special adhesive to allow us to change between them on the resonators experimented channels.

The measurement chain is composed of a white noise generator, signal amplifiers, emission chamber (high frequency tube) and channel with resonators, semi-anechoic reception chamber, acquisition system with microphones and the FFT analyzer. To determine the transmission loss for each experimental model, we have taken into account the measured correction of the main channel without any resonator. The comparative analysis for the 31 measurements is presented in Fig. 7.

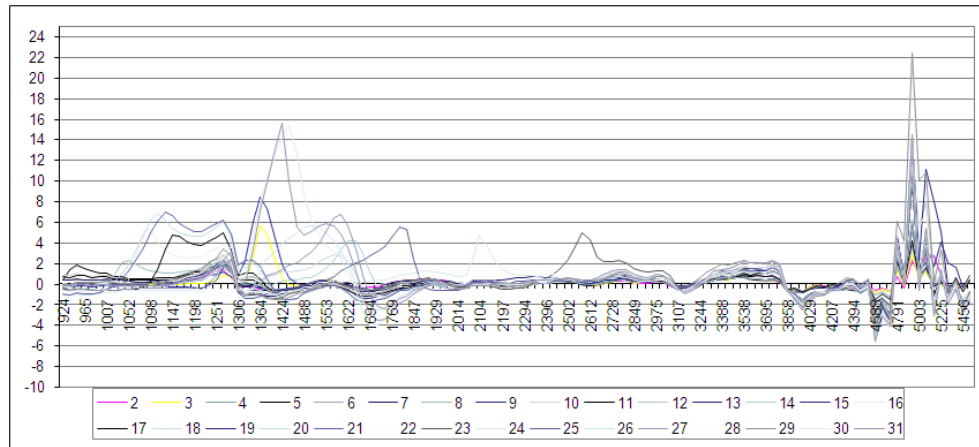


Fig. 7. Comparative analysis of the TL on the frequency range 924Hz – 5656Hz for all the resonator experimented

From Fig. 7, we can conclude that each resonator has different behaviour both in the 1000Hz–1900Hz frequency range and between 4800Hz–5200Hz. We identified the best acting resonators for the 5kHz frequency and we compared them in Fig. 8.

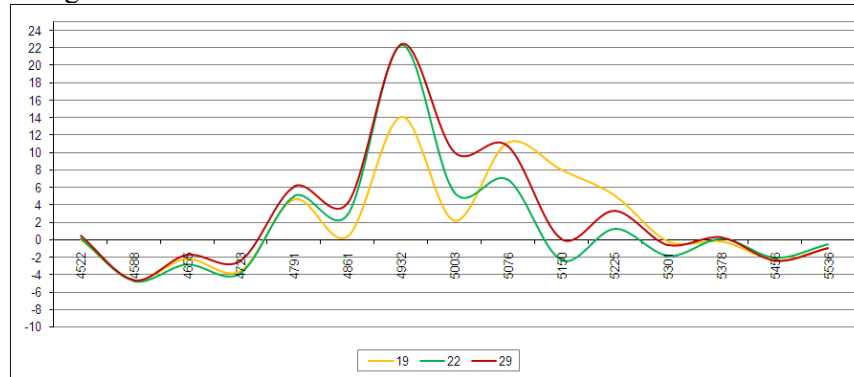


Fig. 8. TL comparative analysis on the 4522Hz – 5538Hz frequency range for the best 3 resonators experimented : model 19 (1x 6mm closed tube), model 22(1x 8mm closed tube) and model 29(2x 8mm closed tubes)

4. Implementation of the resonators for a centrifugal compressor and evaluation of their performances to reduce noise

Having at the base of these researches a national research project – NOVICOM [13] – we choose to implement 8 mm closed tube resonator in the EEISG centrifugal compressor stator geometry.

After implementing the resonators, we performed a new set of measurements in the same measurement points and in the same operating conditions of the centrifugal compressor. The obtained global result is presented in Fig. 9.

Fig. 10 presents the mapping of the noise intensity levels obtained after implementing the resonators for the 5 kHz central frequency and highlights the acoustic phenomenon caused by the 21 rotor blades of the centrifugal compressor, the rotation effect and the fact that the acoustic energy density is increasing in the exhaust area. We can clearly see that the noise levels from 5kHz are significantly reduced after implementing the resonators in the EEISG compressor.

The attenuation on 5 kHz central frequency is about 16,9 dBA sound power level and the global sound power level of the entire compressor case surface is 2,7 dBA (Fig. 11).

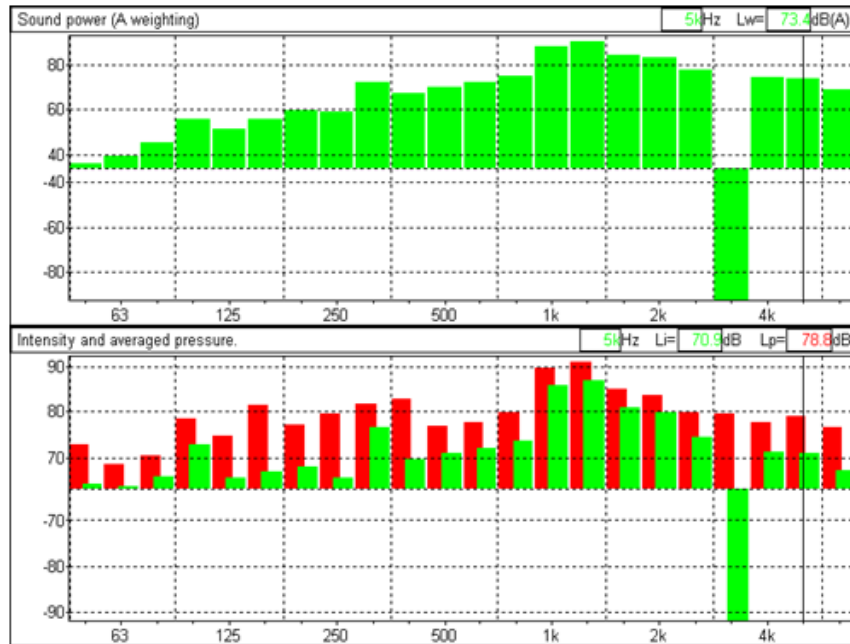


Fig. 9. Global levels obtained by scanning the centrifugal compressor case after implementing the resonators in the compressor geometry

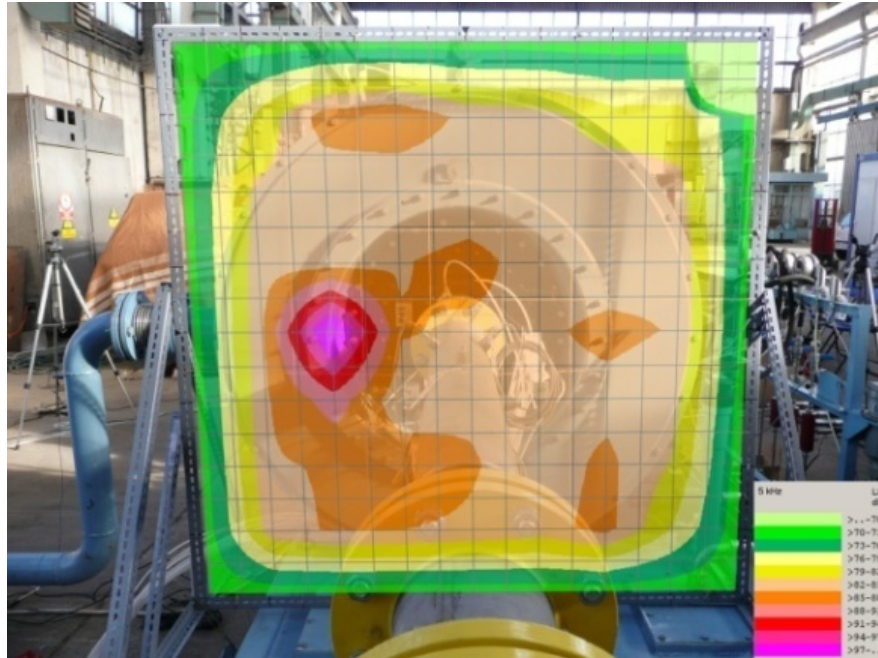


Fig. 10. Noise intensity mapping after implementing the resonators for 5kHz central frequency

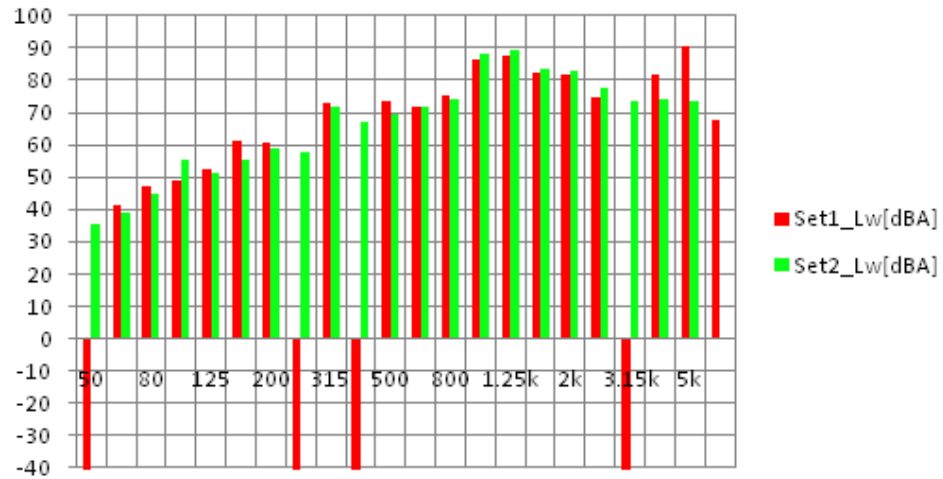


Fig. 11. Acoustic power spectrum comparative analysis (Set1 before implementing the resonators and Set2 after implementing the resonators)

5. Conclusions and further work

The original aspects of this work are:

- the noise reduction at source approach;

- identification of the spectrum frequency which must be reduced, performing noise intensity measurements by scanning 49 surfaces disposed on the centrifugal compressor case with the intensity probe;
- study of the resonators models and establishment of the mathematical models to calculate the transmission loss;
- calculation of the resonators in order to design resonant cavities to act for 5kHz central frequency;
- experiments in the laboratory for 31 resonators models, both Helmholtz and closed tube;
- analysis of the results obtained in the laboratory and choice of the best geometry for the centrifugal compressor studied;
- application of resonant cavities in the centrifugal compressor stator and to obtain significantly noise reduction for a COMOTI centrifugal compressor, about 17 dBA (noise power level) at 5kHz central frequency.
- interpolation of the obtained results of sound intensity to perform the noise mapping of the centrifugal compressor case.

Further, in order to obtain a higher global noise reduction for the EEISG we intend to improve the closed tube resonator geometry in such way as to enlarge the frequency domain in which these are acting. Laboratory experiments must be performed to determine how different geometries of the resonators are interacting between them, without amplify any frequency. Also, in order to increase the efficiency of the resonators different types of treatments must be experimented in the laboratory. These additional researches will be performed in the frame of the NOVIOCM research project.

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